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INVESTIGATION OF ACOUSTIC PROPERTIES OF A RIGID FOAM WITH APPLICATION TO NOISE REDUCTION IN LIGHT AIRCRAFT

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I. SUMMARY

NASA plans to incorporate a rigid, closed-cell foam into the walls of light aircraft cabins to increase their fire resistance. The research program described in this report was undertaken to evaluate the acoustic properties of a rigid, closed-cell foam for use in improving the acoustic environment of light aircraft. The program consisted of three major phases including:

1) Development of a suitable analytic model of sound transmission into an aircraft cabin.
2) Identification of test procedures which appropriately rank order properties which affect sound transmission.
3) Measurement of pertinent properties of materials or constructions which incorporate the foam.

The proposed analytic model agrees well with available data, and reveals that the pertinent properties of an aircraft cabin for sound transmission include: stiffness of cabin walls at low frequencies (as this reflects on impedance of the walls) and cabin wall transmission loss and interior absorption at mid and high frequencies. It was found experimentally that below 315 Hz the foam can contribute substantially to wall stiffness and and sound transmission loss of typical light aircraft cabin construction, and could potentially reduce cabin noise levels

by 3-5 dB in this frequency range at a cost of about 0.2 lb/sq. ft. of treated cabin area. Data on current aircraft reveal that present sound levels in this frequency range produce hearing damage risk with long-term exposure. The foam was found not to have significant sound absorbing properties, but this is not believed to detract significantly from total cabin absorption.

2. A review of measurement techniques which may be used to rank order the acoustic effectiveness of materials or constructions for the various parameters in the model.

3. Sections which present measured data on the NASA foam on constructions involving it and discussions of test methodology.

4. Summary sections which discuss the effectiveness of the NASA foam and suggestions for additional directions of exploration on this problem.

III. SOUND TRANSMISSION INTO AN AIRCRAFT FUSELAGE

There are three principle mechanisms by which sound is transmitted into an aircraft cabin in cruise flight. These mechanisms include:

a) Airborne sound transmitted from propulsive elements (engine inlet, casing and exhaust noise, propeller noise) through the cabin surfaces.

b) Transmission of structure-borne noise from vibrating components by solid paths to surfaces within the cabin which radiate sound.

c) Pressure fluctuations on the cabin surface, due to the flow of turbulent air past the cabin, induce motion in the cabin walls which may then radiate sound.

In modern light unpressurized aircraft, relatively low flight speeds, and good vibration isolation practices for propulsive elements generally mitigate against the second and third noise mechanisms as being major sources, at cruise; so in the following discussions, only the first mechanism will be discussed in detail. Some proof of the above generalizations may be had from considerations of some specific conditions where they are not true. For example, in a decent at idle power and cruise speed, aerodynamic induced noise is generally clearly audible, but at a substantially

reduced cabin sound level in comparison with cruise conditions.

A. An Analytical Model of Aircraft Cabin Noise Reduction

The analytical model for airborne sound transmission into the cabin of a conventional light twin aircraft is based on the following hypotheses:

a) The sound field on the exterior of the cabin is essentially uniform in level and spectrum shape over all exterior surfaces, and when averaged over all surfaces, contains nearly all angles of incidence.

b) The transmitting areas of the cabin surface can be modeled as having an average Transmission Loss (TL) which is determined only by the TL of the component sections weighted by their surface area S.

c) The effective acoustic absorption of the interior of the aircraft is representable as the sum of the products of sabin absorption coefficient times surface area of each absorbing element.

As a consequence of these assumptions, the noise reduction for frequencies well above the first acoustic resonance of the cabin interior is given by

$$NR = \text{the larger of } TL + 10 \text{ Log } \frac{A}{S} \text{ or } 0 \qquad (1)$$

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where $NR = \bar{L}_p(\text{out}) - \bar{L}_p(\text{in})$ is the difference between the space average sound pressure level (SPL) outside the cabin (corrected for pressure doubling at the cabin surface) and the space average SPL inside the cabin.

$TL = 10 \text{ Log } S - 10 \text{ Log } \sum_i S_i \tau_i$ is the area weighted average field incidence transmission loss of the cabin.

$A = \sum_j S_j \bar{\alpha}_j$ is the total sabin absorption in the interior.

S is the total transmitting surface area of the cabin.

It is claimed that Eq. 1 represents an engineering approximation of a lower bound for NR in all frequency ranges above the first acoustic resonance of the cabin interior. Experimental justification for this claim will be presented later in this section.

Below the first acoustic resonance of the cabin interior (typically $f < c/2L$, where c is the velocity of sound in air (at cabin altitude) L is the cabin length), this prediction scheme is not relevant, since this scheme assumes that the volume is multimodal. Another model, which is appropriate at very low frequencies below both the first acoustic resonance of the cabin, and the first resonance of subpanels (or windows) is given by:[1]

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[1] R.H. Lyon, "Noise Reduction of a Rectangular Enclosure with One Flexible Wall", JASA; 35 (1963), p. 1791.

$$NR \geq 20 \text{ Log } 1 + \frac{Z_{walls}}{Z_{volume}} \tag{2}$$

where NR is as defined above.

Z_{walls} is the acoustic impedance of the walls which for frequencies well below the first acoustic resonance is given by (N-sec/m^5)

$$Z_{walls} \simeq \frac{\pi^4}{4^3\omega} \frac{1}{\sum_i S_i/\rho_{si}\omega_{11,i}^2} \leq \frac{\pi^4}{4^3\omega} \frac{\rho_s\omega_{11}^2}{S} \tag{2a}$$

Z_{volume} is the acoustic impedance of the volume given by (well below the first volume resonance) (N-sec/m^5)

$$Z_{volume} \simeq \frac{\rho c^2}{\omega V} \tag{2b}$$

and

ω is the (radian) frequency of interest (Hz)

$S_i;S$ are the are the area of the i element and total cabin, respectively (m^2)

$\rho_{s,i};\rho_s$ are the area mass density of the ith element, and the panel with the lowest resonant frequency, respectively (kg/m^2)

$\omega_{11,i};\omega_{11}$ are the (radian) frequencies of the first panel resonance of the ith element and the element with the lowest natural frequency, respectively.

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ρc is the characteristic resistance of the air in the cabin (406 MKS rayls at STP, 290 MKS rayls at 10,000 ft., std. atmosphere).

c is the velocity of sound for air in the cabin (340 m/sec at STP, 328 m/sec @ 10,000 ft. std. atmosphere)

V is the volume occupied by the air in the cabin (m^3)

Unfortunately, in typical aircraft constructions, the frequency of the first resonance of a typical subpanel is substantially less than the frequency of the first volume resonance, so that there is a frequency regime where the above relationship cannot be used, since the impedance of the walls cannot be simply estimated. In this range, however, motion of the subpanels of the walls is resonant, rather than stiffness controlled, so that statistical techniques may be used to find a first order approximation. The acoustic impedance of the cabin walls may be estimated from the relationship between mean square panel velocity and mean square pressure in an existing sound field[2]:

[2] Noise and Vibration Control, L.L. Beranek, Ed.; p. 301-2 note the changes due to the fact that the panel is assumed to be excited from one side only.

$$|Z_{wall}| = \frac{1}{\left[\sum_i S_i \frac{\langle v_i^2 \rangle}{2\langle p^2 \rangle}\right]^{\frac{1}{2}}} \approx \frac{1}{S}\left[\rho c \frac{4\rho_s \omega^2}{\pi \omega_c \sigma_{rad}}\left(1 + \frac{\rho_s \omega \eta}{2\rho c \sigma_{rad}}\right)\right]^{\frac{1}{2}} \tag{3}$$

where $2\langle p^2 \rangle$ is the mean square pressure on the surface of the fuselage

$\langle v_i^2 \rangle$ is the mean square velocity of the ith panel

ω_c is the critical frequency of the resonant subpanels

σ_{rad} is the radiation efficiency of a subpanel

and other terms are as defined above.

The impedance of the volume should also be modified by a resonance term so that

$$Z'_{volume} = Z_{volume} \frac{\omega_1^{\,2}}{\omega_1^{\,2} - \omega^2}, \quad \omega < \omega_1$$

(where ω_1 is the frequency of the first volume resonance) for frequencies approaching the first volume resonance.

Forming the ratio of wall impedance to volume impedance, we find

$$\frac{Z_{wall}}{Z_{volume}} \approx \frac{2V\omega^2}{S\rho c^2}\left[\frac{\rho c \rho_s}{\pi \omega_c \sigma_{rad}}\left(1 + \frac{\rho_s \omega \eta}{2\rho c \sigma_{rad}}\right)\right]^{\frac{1}{2}}\left(\frac{\omega_1^2 - \omega^2}{\omega_1^{\,2}}\right) \tag{4}$$

for $\omega_{11} < \omega < \omega_1$ (i.e., above first subpanel resonance but below the first acoustic resonance of the cabin).

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As will be seen in a later example, this expression predicts a substantial reduction in wall impedance, as compared with below the first subpanel resonance, as should be expected, resulting in a very low value of cabin noise reduction.

D. Limitations of the Model

It is appropriate to also note some of the anticipated potential limitations of the model and sketch routes for improvement as appropriate. The principle anticipated problems are associated with assuming a spatially uniform acoustic excitation, neglecting flow noise, and determining an appropriate cabin transmission loss. The fact that structureborne noise has been neglected is not considered a severe limitation, since this path is independent of airborne transmission, and techniques are available for predicting this component of noise transmission into an enclosed space.[3]

The uniform excitation assumption interacts with the TL difficulties in that locally high exterior SPL's may be arranged with locally poor TL to produce an actual NR which may be somewhat lower than the lower bound estimate. This difficulty may be correctable if the spatial distribution of SPL can be anticipated, so that this may be paired with the appropriate local TL so that partial power transmission from different areas may be computed and summed.

An additional difficulty with TL is associated with the distribution of sound with angle of incidence. Common TL calculation

[3] See the literature on statistical energy analysis, a brief introduction to which is presented in I.L. Ver and C.I. Holmer, "Interaction of Sound and Solid Structures", Chapter 11, _Noise and Vibration Control_, (L.L. Beranek, ed.), McGraw-Hill, 1971.

and test procedures yield a value which is appropriate for a reverberant field excitation. Fortunately, in many cases where mass law controls the TL, this also approximates the TL at ~60° angle of incidence, which also nearly corresponds to an area-weighted principle angle of incidence on a light twin, where the major sources are probably propeller and engine exhaust noise. Pending more detailed understanding of a particular situation, it is suggested that the field incidence TL represents an appropriate (rather than a conservative) TL estimate.

Neglecting flow noise in the analytic model is the last major hurdle. This was justified on the basis that acoustic sources are expected to dominate the transmission through the hull. As vehicle speeds increase, beyond some point, this cannot be expected to continue to be the case. In commercial jet aircraft, for example, the reverse is believed to be the case.[4] A more complete model including this phenomenon, however, is beyond the scope of this work.

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4. J.F. Wilby, private communication. See for example; W.V. Bhat and J.F. Wilby "Interior Noise Radiated by an Airplane Fuselage Subjected to Turbulent Boundry Layer Excitation and Evaluation of Noise Reduction Treatments", Journal of Sound and Vibration (JSV) (1971) 18 (4) 449-464; J.F. Wilby and F.L. Gloyna "Vibration Measurements of an Airplane Fuselage Structure" J.S.V. (1972) 23 (4), 443-466 and 467-486.

C. Experimental Verification of Cabin Noise Reduction Model

We have available two different sets of data which may be used to test the validity of the analytic model of cabin NR. The first set of data consists of some measurements of sound transmission into and out of small rectangular enclosures, which provide an opportunity to test the analytic expressions of the model directly. The second set of data was taken from a study performed by BBN a number of years ago, on sound transmission into the cabin of a conventional light twin in cruise flight. While the hull construction (in particular the interior trim and window configurations) is not necessarily typical of present-day light aircraft, the study is sufficiently well documented that it will be useful as a test of the model.

The small enclosure evaluation[5] was part of a program to evaluate sound transmission through enclosures, and as such was concerned with the transmission of sound out of, as well as into, an enclosure. Figure 3 presents measured TL data on the wall construction of the enclosure (compared with analytic estimates).

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[5] C.I. Holmer, "Model Tests for Verifying Acoustic Design of an Enclosure for the LM 2500 Module", BBN Report 2343, April 1972. See also C.I. Holmer, "Transmission of Sound Through Enclosures", JASA, 53, (1973), p. 388 (A).

The enclosure consisted of a 1 ft. x 1 ft. x 3 ft. rectangular box of 0.1 in. aluminum walls mounted on a 1 in. x 1 in. box rib frame at all edges, with stiffeners 9 in. o.c. on the four long sides. Absorption in the enclosure was provided by layers of absorbent foam of various thickness. Table 1 provides a tabulation of absorption data within the enclosure measured by the reverberation time technique, for various thicknesses of foam. Figure 4 presents computed noise reduction in accordance with the model. Figures 5 to 8 show a comparison of measured and calculated values of noise reduction and power insertion loss of the enclosure. Noise reduction was measured as the difference between the space average exterior reverberant field SPL and the space average interior SPL.

Power insertion loss of the enclosure is measured as the difference in radiated sound power of a source, with and without the surrounding enclosure. The source for this later evaluation was an array of eight loudspeakers arranged on a cylindrical surface, with various phase relationships. The estimated first acoustic resonance of the enclosed volume is 180 Hz, while the estimated first subpanel (9 in. x 12 in.) resonance (with clamped edges) is 380 Hz, and the first side panel resonance (12 in. x 3 ft. with stiffeners) is approximately 300 Hz. Thus, we do not expect to find a frequency region where the expression with resonant side panels and stiffness

controlled airspace is appropriate. All other aspects of the analytic model are relevant, however; and, thus, the model appears to be well justified. In particular, the model is seen to be appropriate in highly absorbent spaces (c.f. tests with 1 in. lining (Fig. 8) where $\alpha > 0.8$ at and above 1 kHz). Ostensibly one might question using a "diffuse field" hypothesis in this case, but the experimental data suggest that the field is still sufficiently diffuse.

The comparison of sound transmission from inside to outside vs. transmission from outside to inside is also of significance. Operationally, it would appear easier for test purposes to measure fuselage noise reduction by excitation of the interior of the fuselage with a noise source and measure levels outside, than to try and envelope a complete aircraft in a homogeneous noise field.

With this experimental basis for justification of the analytical model, under "ideal conditions", we turn now to an application on some aircraft test data. This data[6] was taken by BBN, with

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[6] L.N. Miller, "Noise Evaluation and Control Recommendations for Cessna ... Aircraft", BBN Report 445, January 1957.

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the sponsorship of Cessna Aircraft Co., under flight conditions on a prototype light twin aircraft. The data were taken at normal cruise (~150 kts IAS) at 10,000 ft. msℓ with unsynchronized engines. Noise level data on the cabin exterior were taken with four flush mounted microphones (Altec BR 150) that incorporated a sintered metal windscreen. Data were taken at four locations, including a) the front windshield, b) side window, c) cabin top (near mid-cabin), and d) at cabin bottom near mid-cabin. Figure 9 shows the variations in exterior octave band SPL vs. position (plotted as deviations from the mean,* for two different flights in the same aircraft. As can be seen, the spread in SPL on exterior cabin surfaces is generally within a ±5dB range except for the front windshield at high frequencies and the top of the cabin at mid-frequencies. For the former case, there is some suspicion of the data because of the improved local NR which occurred when a double window was installed. In the latter case, it was identified that the principle noise source was air flow over a roof mounted radio antenna forward of the microphone. In general, the hypothesis of essentially uniform excitation level over the fuselage seems justifiable. In addition, it should be noted that narrow band (4 Hz) analysis of the tape recorded data indicated that harmonics of engine firing rate and propeller blade rate were clearly distinguishable from 1/2 order to tenth order and controlled the octave band levels through the 300-600 cps bands.

*i.e., average of the four positions

Because of their tonal nature, the interference patterns produced at the surface are probably the principle cause for variation of SPL with position at low and mid frequencies.

Table 2 presents an estimate of cabin absorption. Figure 10 provides a presentation of estimated values of TL plus 3 dB. The three dB correction is applied since the NR comparison will be based on exterior SPL at the cabin surface. TL is shown for the floor construction (.025 AL exterior sheet, 4 in. airspace, .020 AL floor attached by stiffeners ~16 in. o.c.), windows (1/8 in. acrylic) and measured TL data for cabin walls with trim (see Figure 19). From this data, a composite TL is calculated, based on the area weightings indicated in the Figure captions. The computed lower bound for cabin NR is also plotted. Also shown in the figure are estimated values of low frequency NR based on Equations 3 and 4. Figure 11 shows a comparison of calculated and measured cabin noise reduction. The comparison is quite good except in the region of 100-500 hz and above 2 kHz. The fact that experimental NR is high in the low frequency range is attributed to two factors, principally that the excitation is principally tonal in this range, and thus is unlikely to excite panel resonances which limit the NR; and also that the measurements are made in octave bands, so that narrow band dips in NR are more difficult to define. The disagreement at high frequencies is attributed to

an air leak through an air circulation discharge duct in the aft cabin floor and transmission from the aft baggage compartment, which were later identified and treated on other aircraft in this program.

On the whole, however, the agreement is considered to be sufficient so as to justify further use of the analytic model. The pertinent properties of a material or construction, based on the model, are seen to be:

a) At low frequencies (below first cabin resonance) - The contribution of the material to the impedance of a wall construction.

b) At mid and high frequencies (above first cabin resonance) - The contribution of the material to the absorption or transmission loss of a wall construction.

The following section will discuss the merits of small sample test procedures which may be useful for rank ordering performance of materials.

IV. MEASUREMENT TECHNIQUES FOR RANK ORDERING ACOUSTICAL PROPERTIES

There is surprisingly little information in the published literature uncovered in our search of English language literature which discusses the usefulness of sampling or small scale measurement procedures. Because of this, our discussion in this section will be limited in terms of our ability to reach quantitative conclusions regarding the applicability of the test data.

In general, testing of less than full scale components or systems can be broken into two discrete types, namely, small sample tests which expose a portion of an otherwise full scale component to a test environment, and scale model tests which expose the test model of a complete component or system to a scaled test environment. The test program used in this evaluation relies completely on sample tests, but for completeness, we will also briefly consider scale model evaluations. The acoustic properties to be evaluated include contributions to absorption, transmission loss and impedance of constructions.

Sample absorption test procedures are relatively well documented.[7] The two techniques, i.e., tube or reverberation room, give respectively, normal incidence and Sabine absorption coefficients. The coefficient derived from reverberation room measurements is the one which provides data which is appropriate for the analytic model. The normal incidence coefficient, while not providing data which is directly correlated to sabine coefficient, does in fact[8] provide a convenient means for rank ordering performance of similar materials. On this basis, absorption coefficients were measured for small samples of varying density, and compared with measured data on polyurethane foams.

There is one major accepted U.S. standard on laboratory measurement of transmission loss.[9] As its name implies, it was written for application to full scale building components (walls, doors, etc.) and as such is not directly applicable to this problem. Because of this fact, and because there is no substantial literature on this topic of small sample TL measurements, it is appropriate to provide some comment on the expected validity of this type of measurement. The principal problem of small sample TL measurement is associated with the size of the sample with

[7] ASTM C 334-58 "Standard Method of Test for Impedance and Absorption of Acoustical Materials by the Tube Method"
ASTM C 423-66 "Standard Method of Test for Sound Absorption of Acoustical Materials in Reverberation Rooms"
Standards available from ASTM, 1916 Race Street, Philadelphia, PA.

[8] C 384-58, Appendix Al.

[9] ASTM E 90-70 "Standard Recommended Practice for Laboratory Measurement of Airborne Sound Transmission Loss of Building Partitions".

respect to a wavelength of sound, and a bending wavelength in the sample at the lowest frequency of interest. The first consideration arises since the sample is invariably mounted in an aperture in a relatively thick wall in the test facility, which acts as a spatial filter[10], prohibiting the transmission of sound at a number of angles of incidence, depending on frequency, aperture size and depth and sample placement. Recent theoretical work on the TL of apertures[11] has been restricted to considerations of a normal incidence sound field. Unfortunately, much of the sound transmitted through a panel arrives at high angles of incidence, which may be severely restricted in comparison with energy transmitted at normal incidence. This may be accommodated by invoking the restriction that normal incidence TL is changed by say no more than ±1dB. Using the computations of Wilson and Soroka, this criterion is satisfied by all aperture depths for ka = 2 (where a is the aperture radius). For ka = 1.5, however, $\frac{d}{a}$ must be less than 1 (d is aperture depth) and for ka = 1.0, $\frac{d}{a}$ must be in the range 0.5-0.75. Wilson[12] shows data on field incidence TL of a circular aperture with $\frac{d}{a}$ = 0.5 which indicates + 2 dB TL at ka ≃ 2.0 and + 4 dB at ka ∿ 1.0. On the basis of these data

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[10] C.I. Holmer, "Factors Affecting the Comparison of Theoretical and Experimental Sound Transmission Loss of Panels" JASA 49 (1971), p. 88(A).

[11] G.P. Wilson & W.W. Soroka, "Approximation to the Diffraction of Sound by a Circular Aperture in a Rigid Wall of Finite Thickness" JASA 37 (1965), p. 286-297.

[12] G.P. Wilson, "Measurement of the Transmission Loss of a Finite-Depth Aperture", JASA 37 (1965), p. 298-307.

it would appear prudent to limit lowest frequency of interest to $f_i \geq \frac{c}{1.8L}$ (where L is a typical panel length) and $d \leq 4L$, if the TL is intended to be representative of "Field incidence".

The consideration of bending wavelength in the sample is simply to insure that both the sample panel and the full size structure are controlled by similar phenomena in the frequency range of measurement. This applies principally to tests of stiff materials, where it is possible that the sample panel is stiffness controlled at the lowest frequency of interest (i.e., below its first panel resonance) while a full size sample could be several bending wavelengths in dimension at this frequency, and thus be resonant or mass controlled.

Measurements of acoustic impedance of panel constructions have not been described previously in the literature, so that the experience base is not sufficient for us to generalize upon. The concept would appear to permit improved understanding of the response of structures to low frequency sound. It should be noted that application of this technique to lightweight

structures requires lightweight motion transducers which have only recently become available. The technique consists of measuring local exciting pressure and resultant panel velocity, so as to provide information for computing the ratio of $Z = \frac{\langle p \rangle}{\langle v\ \text{normal} \rangle}$ vs frequency. Application of the technique requires the knowledge that the measured velocity is in fact due to the local pressure (i.e., that the surface responds locally) in order for the results to be successfully interpreted. Light skinned, ribbed structures, such as aircraft cabins, are expected to fulfill this requirement easily.

Scale models are another useful technique for determining experimentally the answers to complex acoustical questions. The principle of the scale model is that when a system is modeled with a scale factor S (i.e., $L_{model} = S \times L_{full\ scale}$) then all dynamic characteristics are preserved at the frequency, $f_{model} = \frac{1}{S} \times f_{full\ scale}$. The principal experimental problems are associated with modeling absorption (here flow resistance of the model porous absorber should equal the flow resistance in the full scale absorber) and system damping (i.e., system loss factor in model at scale frequency should equal system

loss factor of the full scale item at normal frequencies). The principal type of difficulty which arises is that of finding suitable materials and construction techniques which permit simulation of the absorption and damping. An additional limitation of modeling is that the model must remain sufficiently large to permit accurate construction in detail. The advantages of modeling are that generally, testing of complex systems can frequently be done more quickly and with less total cost that full scale tests, particularly if the subject does not already exist in full scale.

V. RESULTS OF ABSORPTION MEASUREMENTS ON FOAM

Absorption measurements were made on each of the samples provided by NASA using the B&K impedance tube apparatus in accordance with ASTM C 384-58. A schematic sketch of this apparatus is shown in Figure 12. In operation, the loudspeaker is excited by a tone, which excites the tube. The sum of the direct sound wave from the speaker, with the first reflection from the sample under test, produces a standing wave pattern in the tube. A plot of the sound pressure level (SPL) vs. position in the tube for some relatively high frequency is also shown in Figure 12. The difference between the peak and minimum SPL (called the Standing Wave Ratio -- SWR) is a direct measure of the absorption coefficient of the sample. Absorption coefficient was read directly from an appropriate scale of the spectrometer. The appropriate filter on the spectrometer was used to improve signal to noise ratio. Due to cross resonance interference, a single tube cannot adequately span the frequency range of interest. Two tubes, one 10 cm. in diameter and one 3 cm. in diameter, are used. The large tube is useful over the range of 125 Hz - 1600 Hz. The small tube is useful for the range 1000 Hz - 6300 Hz.

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As a check of the apparatus and technique, a test run was made on a sample polyurethane foam, which has demonstrated good reproducibility. The data from this check are plotted in Figure 13. The plot is of normal incidence absorption coefficient vs. frequency, both on a log scale. Absorption is presented on a log scale since the performance of an absorptive material is proportional to 10 Log α. Also, errors in the measurement are more nearly of the form $\pm 10 \text{Log} \Delta\alpha$ than of the form $\pm \Delta\alpha$.

The comparison indicates that the samples measured are nominally within ±1dB of the manufacturer's data, indicating good absolute reproducibility of the test setup.

Figures 14 and 15 show the measured absorption of the samples provided by NASA. The samples provide substantially less absorption than a "good" absorptive material (such as the polyurethane foam), and there does not seem to be any discernible trend from sample to sample over the entire frequency range. Also shown in these figures is the lower limit of measurable absorption for this apparatus, which has been taken as the apparent measured absorption of the steel termination alone.

A detailed inspection of the samples measured suggests that the mid- and high-frequency absorption coefficients appeared

to correlate with average surface roughness or surface damage (i.e., slight crushing of foam cells on sample surface).

While there was little opportunity to quantify this conclusion, or to investigate other mechanisms for this behavior, the hypothesis appears to be reasonable in view of the apparent closed cell nature of the foam, and does provide a mechanism for explaining the non-systematic variations which were observed.

VI. TRANSMISSION LOSS MEASUREMENTS ON TYPICAL CABIN CONSTRUCTION

The TL measurements were undertaken in a research test facility at BBN which has been described in detail elsewhere[13].

A. Experimental Procedure

Figure 1 shows a plan view of the test suite, while Figure 2 shows a schematic diagram of the TL test facility instrumentation. For testing, a sample was mounted in a 3 ft. x 3 ft. framed opening in the modular filler wall with the "exterior face" toward the source room (lab B). During a test, Lab B is excited with a broad-band noise signal (derived from an amplified pink noise signal fed to an Acoustics Research LST loudspeaker system). Level of this signal is determined to be sufficient so that the signal transmitted to Lab A is easily

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13. C.I. Holmer and D.W. Andersen, "New Reverberant Room Acoustic Test Facility". JASA 53 (1973), p. 302 (A)

measured above the existing ambient. The one-third octave band space average sound pressure level is determined from a four-microphone array in each Lab. From these spectra, the Noise Reduction (NR) as a function of frequency is determined according to the equation:

$$NR(f) = \overline{SPL}_{Lab\ B}(f) - \overline{SPL}_{Lab\ A}(f) \qquad (5)$$

At this point, the noise reduction for the sample is compared with the measured noise reduction for the solid filler wall to determine the magnitude and correct for the transmission through the filler wall. No correction was made, unless the sample NR exceed the filler wall NR by at least 3 dB. (Data not meeting this criterion is not reported.) Corrections typically were only required in the last 2 or 3 one-third octave bands reported. The transmission loss is determined from the NR according to the expression:

$$R(f) = NR(f) + 10 \text{ Log } \frac{S}{A(f)} \qquad (6)$$

where: R(f) is the transmission loss at the frequency f

NR(f) is the noise reduction

S is the sample surface area

A(f) is the sabin absorption of the receiving room (Lab A)

determined from $A(f) = \frac{60\ V}{cT_{60}(f)}$ (7)

where: V is receiving room volume

c is the speed of sound

$T_{60}(f)$ is the room reverberation time

A and S are in consistent units. This measurement and analysis procedure corresponds explicitly to ASTM E90-70 "Standard Practice for Laboratory Measurement of Airborne Sound Transmission Loss", except for sample size. Figure 16 presents a comparison of TL data generated in this Lab of a 3'x3' sample with TL data generated in several other labs[14] on a 22 gauge galvanized steel sample of various dimensions. TL of this sample is essentially determined by mass law over the entire frequency range. The other laboratories represented include: National Research Council, Ottawa; Riverbank Acoustical Laboratories, Owens-Corning Fiberglass Corp., National Gypsum Co., and Lord Manufacturing Co. All samples except that from Lord Manufacturing Co. were substantially larger than the sample used here. This suggests that data with this sample size and aperture

14. C.I. Holmer, "The Lord Acoustic Test Facility", JASA 44 (1968), p. 359(A).

depth is not significantly affected above 200 Hz. This result is consistent with the suggestions of Section IV of this report.

B. Experimental Data

The series of transmission loss measurements included the following samples:

1. .032 in. Aluminum Sheet
2. Basic panel no. 1
3. Basic panel no. 2
4. Basic panel no. 3 with NASA applied foam (∿3.5 lb/ft^3 density 1½ in. thick)
5. Basic panel no. 1 plus glass fiber insulation
6. Basic panel no. 1 plus glass fiber plus "interior trim". (Three mountings for interior trim)
7. Basic panel no. 3 with NASA foam plus "interior trim".

Figures 17a to 17c depict the construction of the "basic panel". Figure 17d presents the TL data on samples 1-4.
Figure 18 shows the results of measurements on the conventional ribbed panel with several forms of simulated "interior trim". The addition of 1 in. of a low density fine fiberglass mat (Owens Corning type 702, ∿1.3 lb/cu.ft., density) provides a progressively increasing attenuation at higher frequencies. It is expected that the addition of a thin covering such as

solid or perforated vinyl film up to 4 mil thick will not substantially change this result below 4 or 5 kHz. Discussion with Cessna indicated that typical practice for interior trim below windows consists of a membrane of "ruggedized" aluminum sheet covered with a "leatherette" or other decorative facing (total weight 0.30-0.35 lb/ft^2) attached to the fuselage whenever or however possible. To simulate this we attached an 0.025 in. thick aluminum sheet to the interior of the panel either continuously along the top of the major ribs (hat sections of Figure 16) or at points on the ribs ∿18 in. o.c. (three points per rib on these samples). Attachment was by means of ½ in. wide double sided scotch tape. While tape is not typical of actual construction practice, it is believed to be dynamically representative of rigid attachments such as rivets, snap fasteners, clips, etc. As can be seen, the improvement in TL is determined in part by the density of attachment points. The two test configurations with attachments were selected to be representative of the probably extremes of possible installation practices. Thus, the TL of fuselage sections incorporating this style of single layer massive interior trim is expected to be bounded by these results.

Figure 19 shows the results of measurements of the same "interior trim" configurations applied to the NASA treated panel.

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Note that the foam completely filled the space between major ribs so that no room was available for glass fiber. Of significance is that improvement in TL does not begin until 1250 Hz or so, and is independent of density of attachment points in the region of measurement.

C. Discussion of Transmission Loss Data

The addition of ribbing and rivets increased total panel weight by approximately 100%, giving an expected 6 dB increase according to simple mass law expectations. Since the weight is concentrated over less than 50% of the panel area, this would lead to inferring a maximum TL increase of ~3 dB above a typical subpanel resonance. A straightforward calculation indicates (assuming simple supported edges along rivet lines) that the first resonance of most subpanels lies in the 100 Hz one-third octave band or lower. The principal role of the ribs is expected to be to increase the local mass in the area covered by ribs, reducing the effective transmitting area. A notable exception occurs when the attachments undergo resonant deformation. Principal ribs, for example, have 1.6 in. side walls, corresponding to a half bending wavelength at ~1 kHz. The unimproved TL at this frequency is probably due to this effect. The addition of the foam increases both the mass and stiffness of

the subpanels, resulting in higher subpanel resonant frequencies. The resulting system has more uniform mass and stiffness leading to more uniform response to the soundfield and response that is controlled by the total mass per unit area of the panel. This form of response is maintained up to the frequency regime where rib motion again controls the radiation (i.e., ~1000 Hz).

Figure 20 presents a comparison of calculated[15] and measured improvement in TL due to the addition of simulated "interior trim" to the untreated and treated panels. The theory of this construction is that the initial frequency and slope is determined by a resonance of the mass of the layer against the stiffness of the contained airspace. Theory predicts that

$$f_o \approx \sqrt{\frac{23}{md}} \qquad (8)$$

where: m is the mass/unit area of the attached layer (kg/m^2)

d is the depth of the airspace between the basic radiating structure and the attached layer (m).

The initial slope of the ΔTL curve is 12 dB per octave. The maximum improvement is determined by reradiation from the attachment areas. The discrepancy between theory and calculation

15. Noise & Vibration Control, L.L. Beranek, ed., McGraw-Hill, 1971, pp. 320-325

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in the vicinity of 1 kHz is again attributable to the hypothesized large motion of the rib surfaces in this frequency range. This is confirmed by the data with no attachments.

For the case of the interior trim attached to the NASA treated sample, the calculated depth of airspace to produce the observed f_o is 1/32 in. This is consistent with the observation that the foam is closed cell, and that the foam was applied to the height of the ribs leaving no substantial airspace between it and the trim panel.

VII. RESULTS OF IMPEDANCE MEASUREMENTS ON TYPICAL CABIN CONSTRUCTIONS

The specific acoustic impedance data reported here was taken with the panel installed as for TL measurements. The measurement is based on the relationship[16]

$$|Z_s| = \frac{|p|}{|u|}$$

where $|Z_s|$ is the magnitude of specific Acoustic Impedance ($N\text{-}sec/m^3$)

$|p|$ is the mean acoustic pressure at the point of observation

$|u|$ is the mean normal velocity of the panel at the point of observation

16. P.A. Franken, "The Behavior of Sound Waves", Chapter 1 of Noise and Vibration Control (L.L. Beranek, ed.) McGraw-Hill (1971) p. 23.

This impedance differs from that discussed in Section III by factor of $(\text{area})^{-1}$. For the measurement, a panel was excited by a sound field as for a TL measurement (with the panel "exterior" surface nominally flush with the filler wall). SPL and acceleration level are sensed at a point on the panel with calibrated ½ in. microphone system and a calibrated accelerometer system, and analyzed in 1/3 octave bands using the instrumentation system of Figure 2. Impedance level is computed from

$$20 \text{ Log } Z_s(f) = SPL(f) - AL(f) + 20 \text{ Log } f - 98 \text{ dB} \qquad (9)$$

where $Z_s(f)$ is the specific acoustic impedance ($N\text{-sec}/m^3$)

SPL(f) is the sound pressure level (dB re $2 \cdot 10^{-5}$ N/m^2)

AL(f) is the acceleration level (dB re 1g)

f(f) is the frequency of interest

The results of such measurements are shown in Figures 21 and 22 for the basic panel and the NASA treated panel. It is evident that the NASA treated panel has a substantially higher impedance (lower response to exciting sound fields) than the basic panel. The differences in response are on the order of 10 dB from 63 Hz - 125 Hz. Comparisons of the data below 63 Hz are subject to question since the first panel resonance

for this size ribbed panel is on the order of 63 Hz, and would be expected to be lower for larger panels, which will alter the comparison. Figure 21 shows a calculated value of Z_s for the largest subpanel using an appropriately modified form of equation 3 (i.e., $Z_s = S\ Z_{wall}$). The surprising result that the estimate based on eq. 3 is approximately 10 dB above the lowest observed values is not yet understood.

VIII. DISCUSSION OF RESULTS

The results of the absorption tests strongly indicate that the foam provides little useful absorption in the frequency range from 125 Hz - 2 kHz, in comparison to that provided by porous materials. This is attributed to the closed cell configuration of the foam, and is not likely to be altered unless the cell structure is changed to substantially open cell configuration. This may or may not be feasible in light of the impact that this would have on principle non-acoustic properties, and will not be considered further.

The dramatic improvement in TL and impedance which occurred when this material was combined with a conventional aircraft cabin construction is certainly worth further consideration.

Figure 23 presents a compilation of data[17] on interior SPL at normal cruise from a number of current model light twin aircraft, compared with two hearing damage criteria. A second criterion, for speech interference should also be considered. The speech interference level (PSIL), consisting of the arithmetic average of the octave band levels at 500, 1000 and 2000 Hz provides one such metric. A PSIL value of 77 dB, for example, permits barely acceptable communication for distances of about ½ ft., using a raised voice. On these bases, the noise levels in the 125 and 250 Hz octave bands are seen to be excessive because of potential hearing damage risk while the noise levels at mid-frequencies generate a need for high levels of voice effort to permit good communication in the cabin. Problems with radio communication generated by these noise levels are much more difficult to assess, in general, due to the wide range of microphone and audio equipment used (i.e., throat or boom microphones vs. hand-held, and cabin loudspeaker vs. headset). Nevertheless, it is clear that techniques for providing up to 5 or 8 dB of noise reduction at low frequencies and 3 to 5 dB at mid-frequencies should certainly be of interest. Comparison of figures 18 and 19 shows that the foam as applied can potentially provide an average of 4 dB increased TL (and thus

17. J.V. Tobias, "Noise in Light Twin-Engine Aircraft" Sound and Vibration 3, 9 (September 1969) p. 16-19 Figure C3

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potentially also in NR) in the frequency range of 100-250 Hz. This increase is corroborated by observed increases in panel impedance in this frequency range. The principal means by which this was achieved is believed to be through stiffening and mass addition to the larger subpanels so that the total weight of the panel, including the stiffeners, was active in producing the mass law TL which was observed. This is inferred from a comparison of treated and untreated panels (Figure 17), where the increase in TL with the foam is substantially greater than the added weight alone would indicate.

It is believed that a similar improvement can be obtained with thinner treatments of higher density foams with constant added weight. The principal reason for doing this is to create space between the foam surface and the interior trim so as to reduce the double wall resonance frequency. With this treatment, (i.e., foam, 3/4 in. airspace, glass fiber blanket, and interior trim) it is believed that a TL curve which is comparable with the basic panel plus standard trim can be achieved. In addition, filling the cavities behind the major ribs, could sufficiently control the rib resonance, so that an improvement of as much as 5 dB in cabin NR at 1000 Hz could be achieved. A caution is in order at this point, since it is likely that the TL of typical

thickness windows may limit the cabin NR in this frequency range, such that large increases in cabin wall TL will not produce significant changes in cabin NR. (See Figure 10.)

At this point, the most appropriate next step would appear to lie in the direction of determining that construction which optimizes cabin TL improvement vs. added weight.

IX. CONCLUSIONS AND RECOMMENDATIONS

A. Regarding the NASA Foam.

The absorption coefficient of the NASA foam is not significant enough so as to be of major interest as an acoustical material. However, other than the effective volume occupied displacing other materials, the foam will not detract from cabin interior absorption. The stiffness of the foam is significant enough so that constructions may be found which will provide a 3-5 dB improvement in cabin NR at low frequencies (63 - 250 Hz) and simultaneously provides a TL at mid and high frequencies which is no less than that observed in present constructions, with a weight cost of about 0.2 lb./ft.^{2}. It is suggested that the next appropriate task is an experimental study of potential constructions which could meet these requirements, prior to installation on an aircraft.

B. Regarding Other Aspects of This Program.

An analytic model of fuselage noise reduction has been proposed which appears justified, based on confirmation with limited experimental data. Based on this model, it appears that the simple

architectural acoustics concepts of Transmission Loss and Sabine Absorption are adequate for determining a lower bound of performance within engineering accuracy. More complex descriptions have been presented for the frequency regimes below the first cabin acoustic resonance, and first wall structural resonance. The description for the frequency range between first structural first acoustic resonance is still subject to more rigorous experimental verification.

We have noted that an important hypothesis of the proposed model for cabin NR is that the sound field is essentially uniform over the transmitting area. An experimental model study to verify the accuracy of this hypothesis for light aircraft, explore the role of diffraction in creating this situation, and study the implications of non-uniform excitation would be a useful contribution to the state of the art in the general understanding of sound transmission into enclosures in a free-field environment. This, of course, has much broader implications than the area of light aircraft.

We have proposed a test procedure for evaluating specific acoustic impedance of structures. This approach may be of more general use in evaluating excitation of complex structures by sound. This

might most favorably be advanced by a study which generates a larger experimental base of experience with the technique as well as additional analytic studies relating the experimental data which is developed.

A final area of note which surfaced in this study is based on the agreement between calculated and measured performance improvement in performance for some double wall constructions (see Fig. 20). This agreement is believed to justify an expanded experimental and analytical study of TL of double wall structures, which may provide a basis for a design guide for TL of double wall aerospace constructions.

X. ACKNOWLEDGEMENT

The invaluable assistance of Douglas W. Andersen and Matthew N. Rubin in performing the measurements reported here is gratefully acknowledged. Numerous members of the staff of BBN contributed to our progress through informal discussion. In addition, I would like to thank Cessna Aircraft Co. for permission to use data from their report, as well as helpful discussions of aircraft construction.

TABLE 1

MEASURED AND CALCULATED SABINE
ABSORPTION FOR ENCLOSURE MODEL

	Sabine Absorption ($S\bar{\alpha}$) frequency							
Unlined Enclosure	125	250	500	1k	2k	4k	8k	16k
Measured $S\bar{\alpha}$	.5	.5	.8	.8	.8	1.1	1.5	1.8
$10 \mathrm{Log} \frac{A_{meas}}{S}$ (dB)	-14.5	-14.5	-12.5	-12.5	-12.5	-11	-10	-9
1/4 in. lining								
measured $S\bar{\alpha}$	1.3	1.0	1.5	3.0	4.0	5.1	9	9
calculated $S\bar{\alpha}$	1.6	2.3	3.4	2.9	3.4	6.0	8.8	-
$10 \mathrm{Log} \frac{A_{meas}}{S}$ (dB)	-10	-11½	-10	-7	-5½	-4½	-2	-2
1/2 in. lining								
measured $S\bar{\alpha}$	1.7	1.7	1.1	5.0	6.1	9	14	14
calculated $S\bar{\alpha}$	1.9	2.6	4.5	4.5	6.0	9.8	11.4	-
$10 \mathrm{Log} \frac{A_{meas}}{S}$ (dB)	-9	-9	-11	-4½	-3½	-2	0	0
1 in. lining								
measured $S\bar{\alpha}$	2.7	2.8	2.7	11	13	13	24	24
calculated $S\bar{\alpha}$	2.6	4.5	5.8	7.0	11.5	12.7	14	-
$10 \mathrm{Log} \frac{A_{meas}}{S}$ (dB)	-7	-7	-7	-1	-½	-½	+2	+2

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TABLE 2

Estimate of absorption of a twin engine, unpressurized light aircraft.

Cabin interior dimensions:

~1.6 m.dia. x 2.0 m. long.

$$f_{001} = \frac{340}{4} = 85 \text{ Hz}$$

I. Absorption Estimate:

Absorbing materials:

a) 1/2 in. glass fiber interior treatment with impervious vinyl covering.
Treatment covers 1/3 of interior surfaces.

frequency	125	250	500	1k	2k	4k
α*	0.09	0.18	0.38	0.9	0.4	0.2

b) Leather (or vinyl) covered seating (six seats occupying total of 2.8 sq. meters including leg room).

frequency	125	250	500	1k	2k	4k
A_{seats}**	1.2	1.5	1.7	1.7	1.6	1.4

(A essentially unchanged whether occupied or not.)

c) Ceiling: 1.6 m. x 2 m. 1/2 in. glass fiber covered with porous cloth.

d) Floor: pile carpet with no pad ~1.6 m. x 2 m. on .020 AL with 4 in. airspace behind.

* Based on scaling data for 1" AEROCORE covered with 4 mil. vinyl film

** Dimensions of metric sabins

Total interior absorption.

	125	250	500	1k	2k	4k
A_{walls}	.36	.8	1.7	3.9	1.6	.9
A_{seats}	1.2	1.5	1.7	1.7	1.6	1.4
$A_{ceiling}$	.15	.5	.9	2.2	2.5	2.2
A_{floor}	1.0	.5	.4	.3	.75	1.3
A_{total}	2.7	3.3	4.7	8.1	6.5	5.8
10 Log A/S	-5.5	-4.5	-3	-1	-2	-2

Total surface area (excluding floor area, aft (baggage compartment) bulkhead, and instrument panel) ~9.6 sq. meters.

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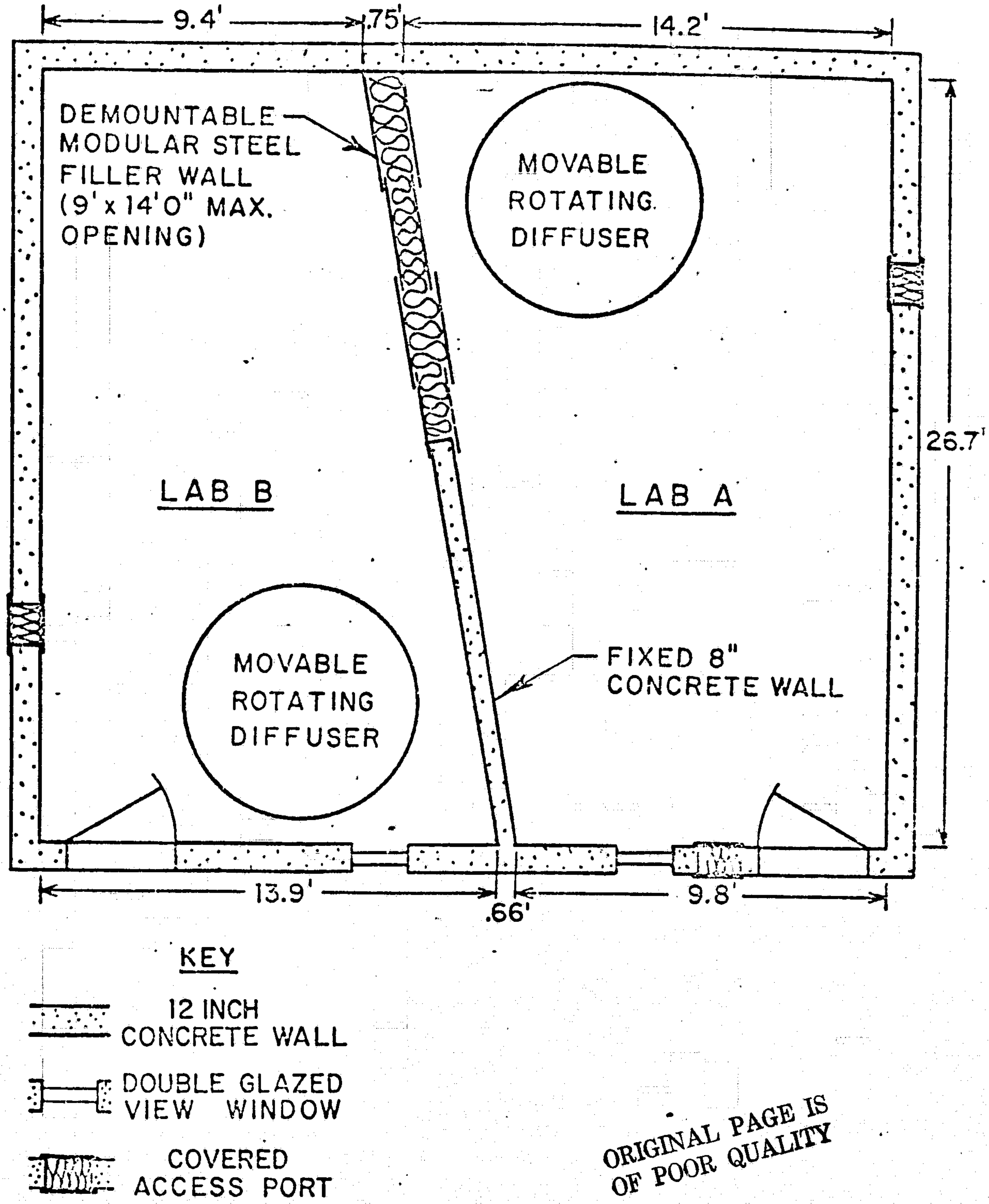

FIG. 1. TEST FACILITY PLAN

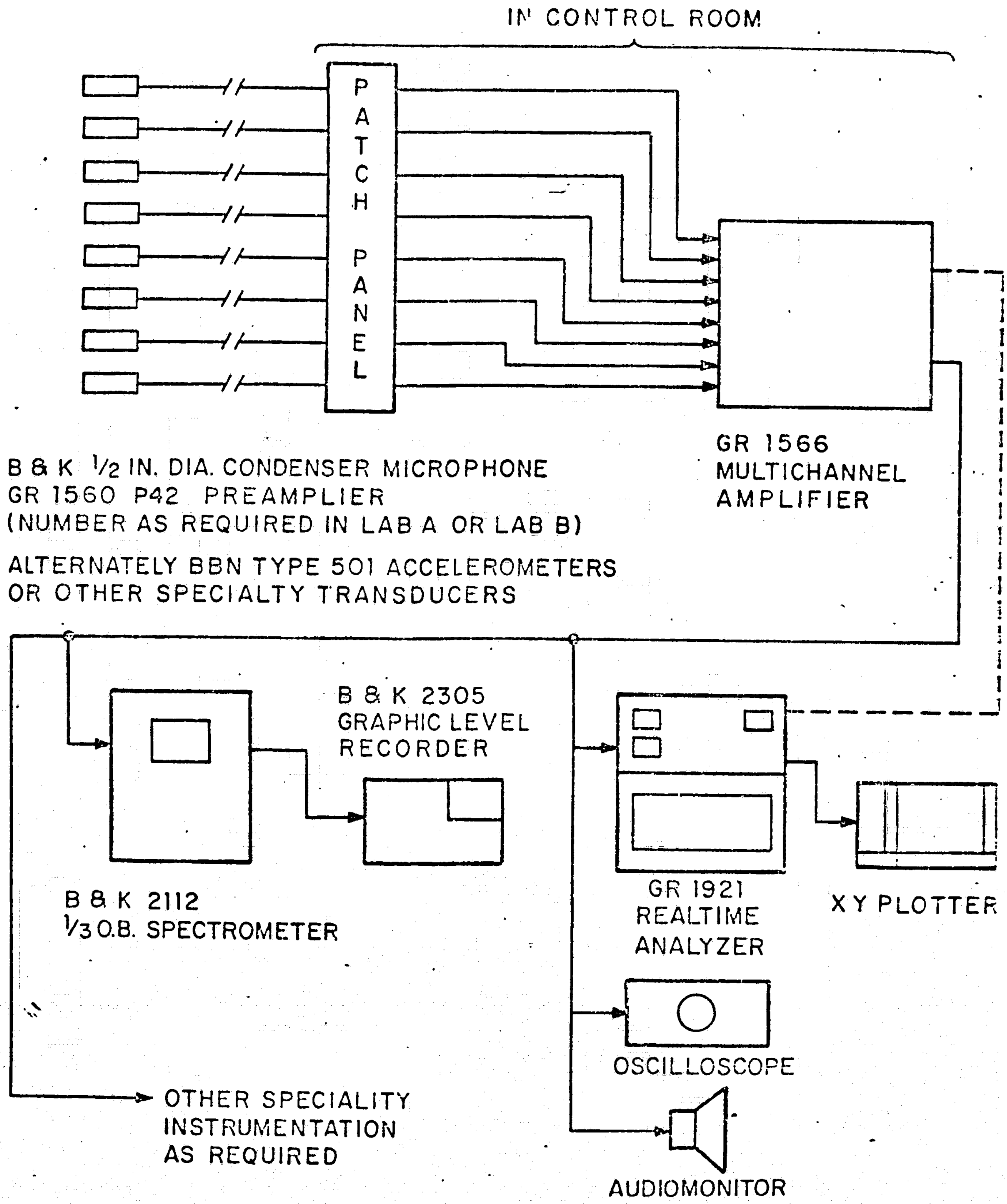

FIG. 2. TEST FACILITY INSTRUMENTATION

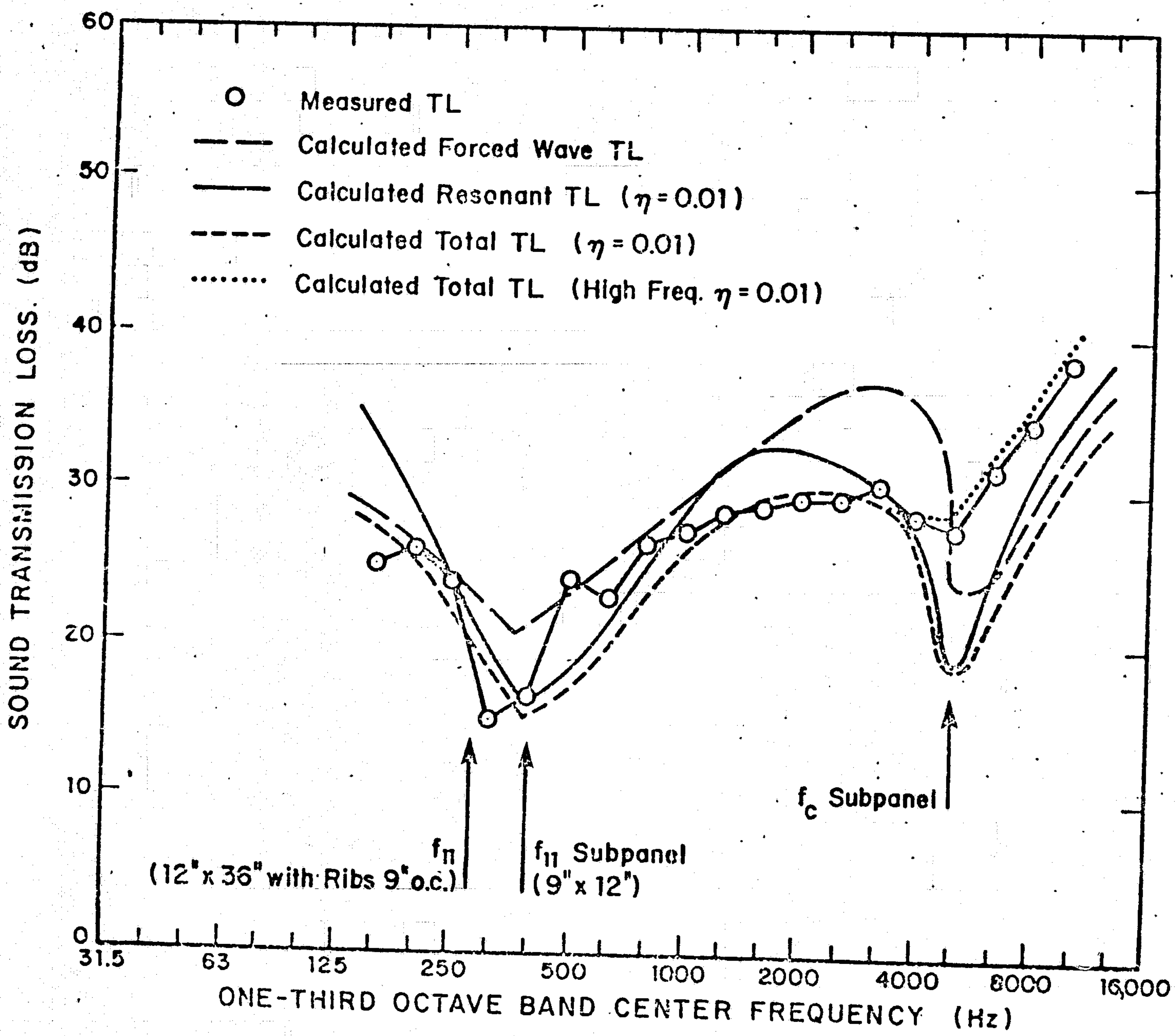

FIG. 3. SOUND TRANSMISSION LOSS OF ENCLOSURE PANEL

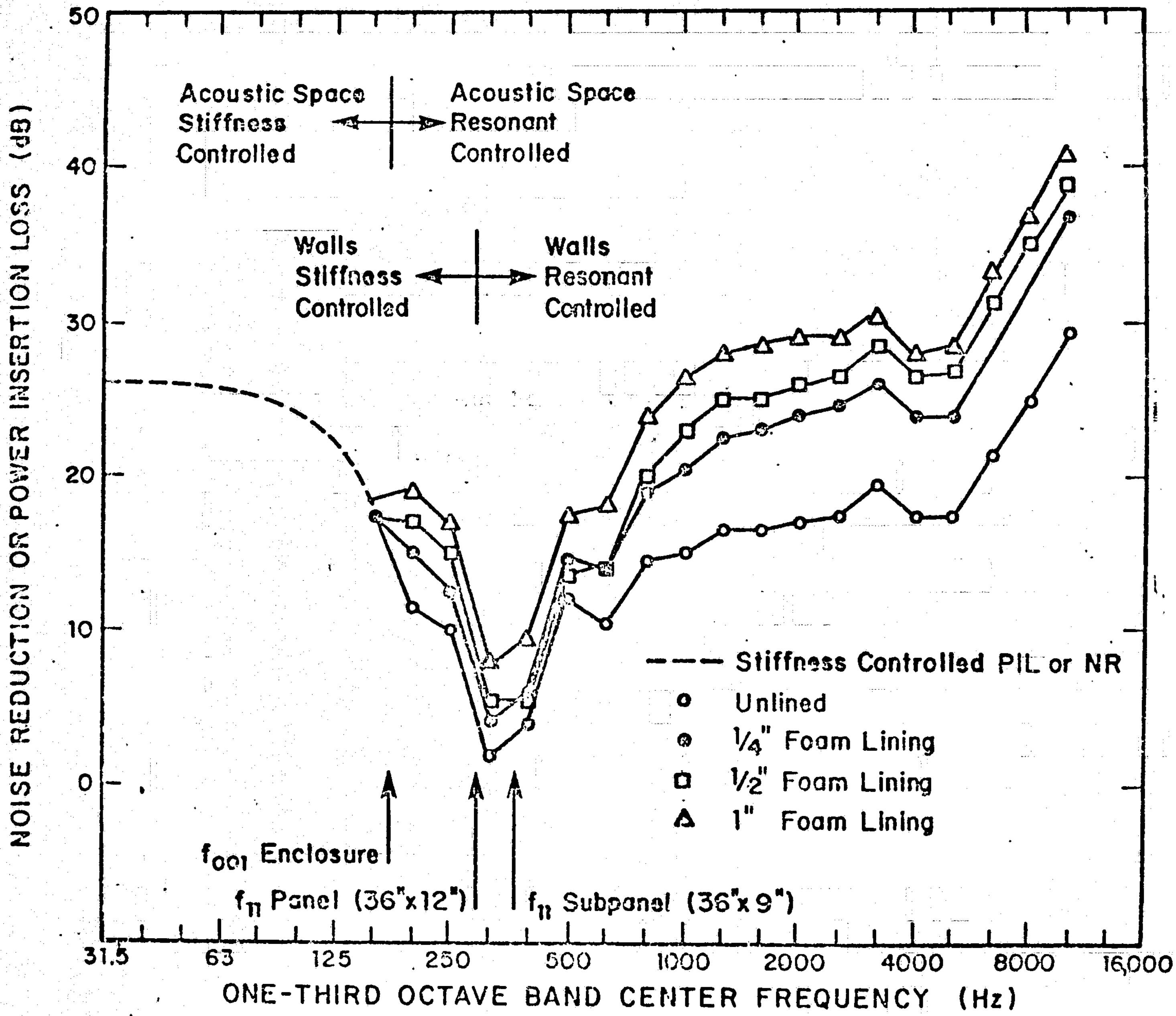

FIG. 4. COMPUTED NR OR RRP (USING MEASURED TL AND A) FOR MODEL ENCLOSURE

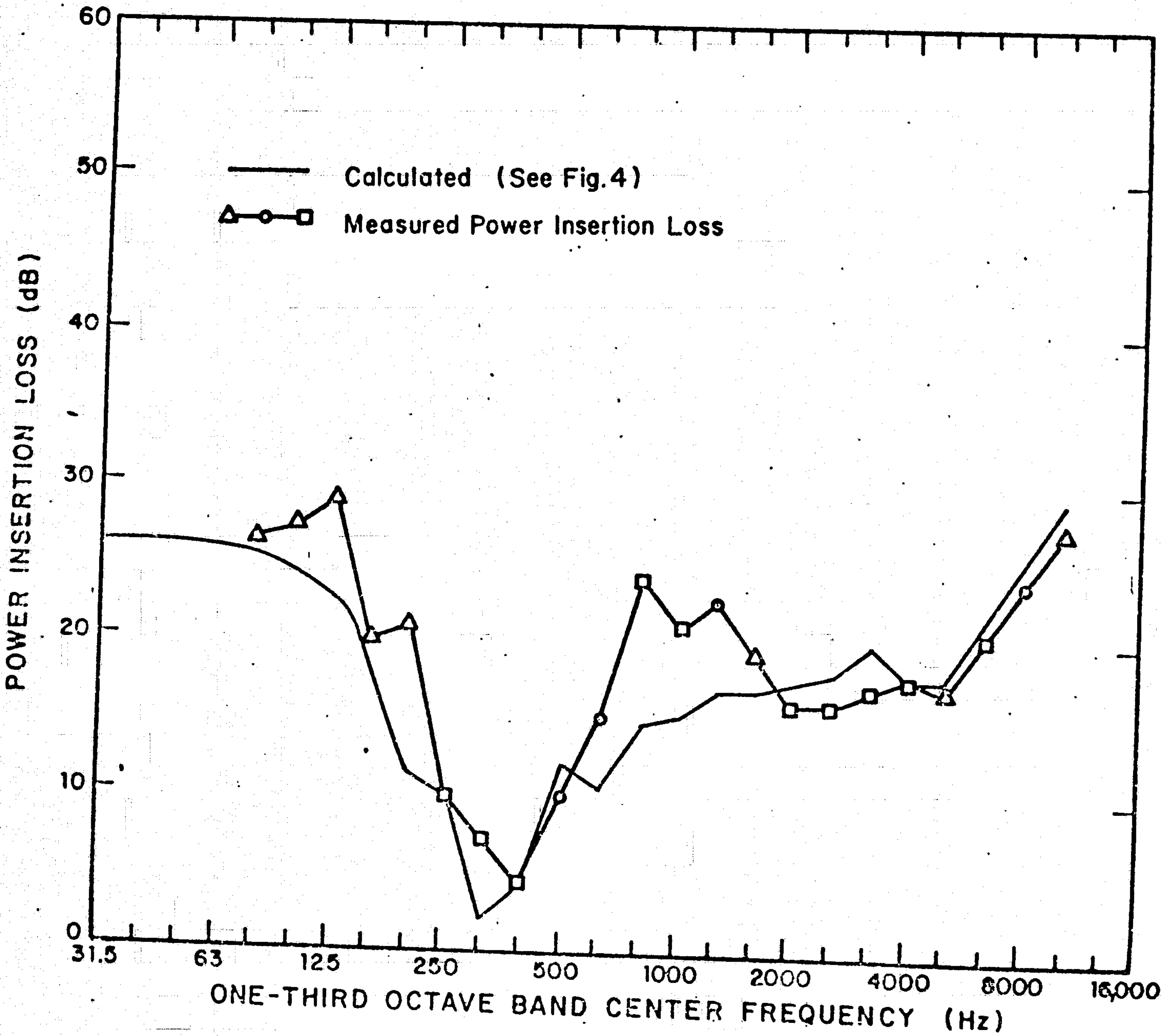

FIG. 5. CALCULATED VS MEASURED ENCLOSURE PERFORMANCE WITH NO ABSORPTION

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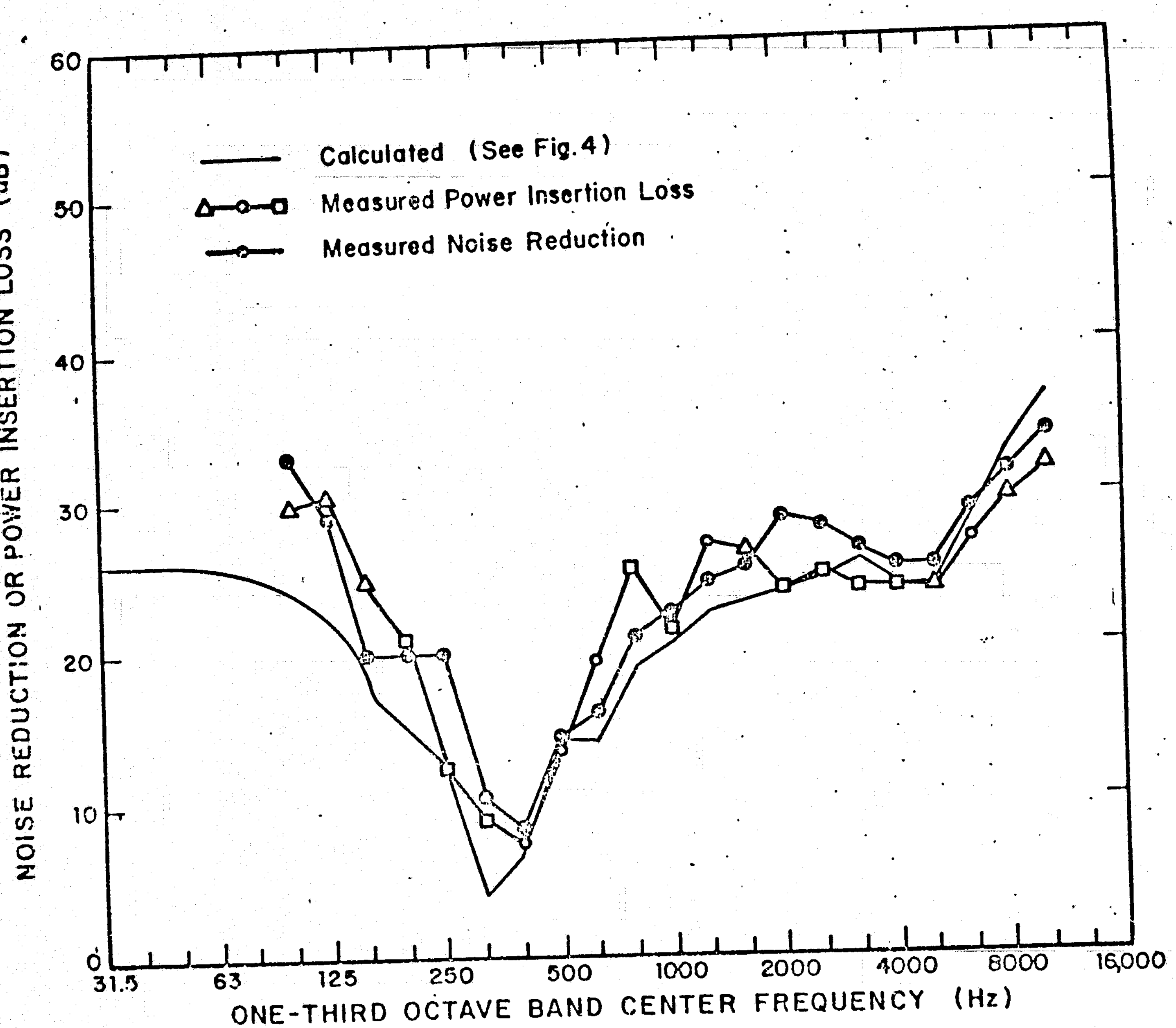

FIG. 6. CALCULATED VS MEASURED ENCLOSURE PERFORMANCE WITH 1/4-IN. ABSORPTIVE LINING

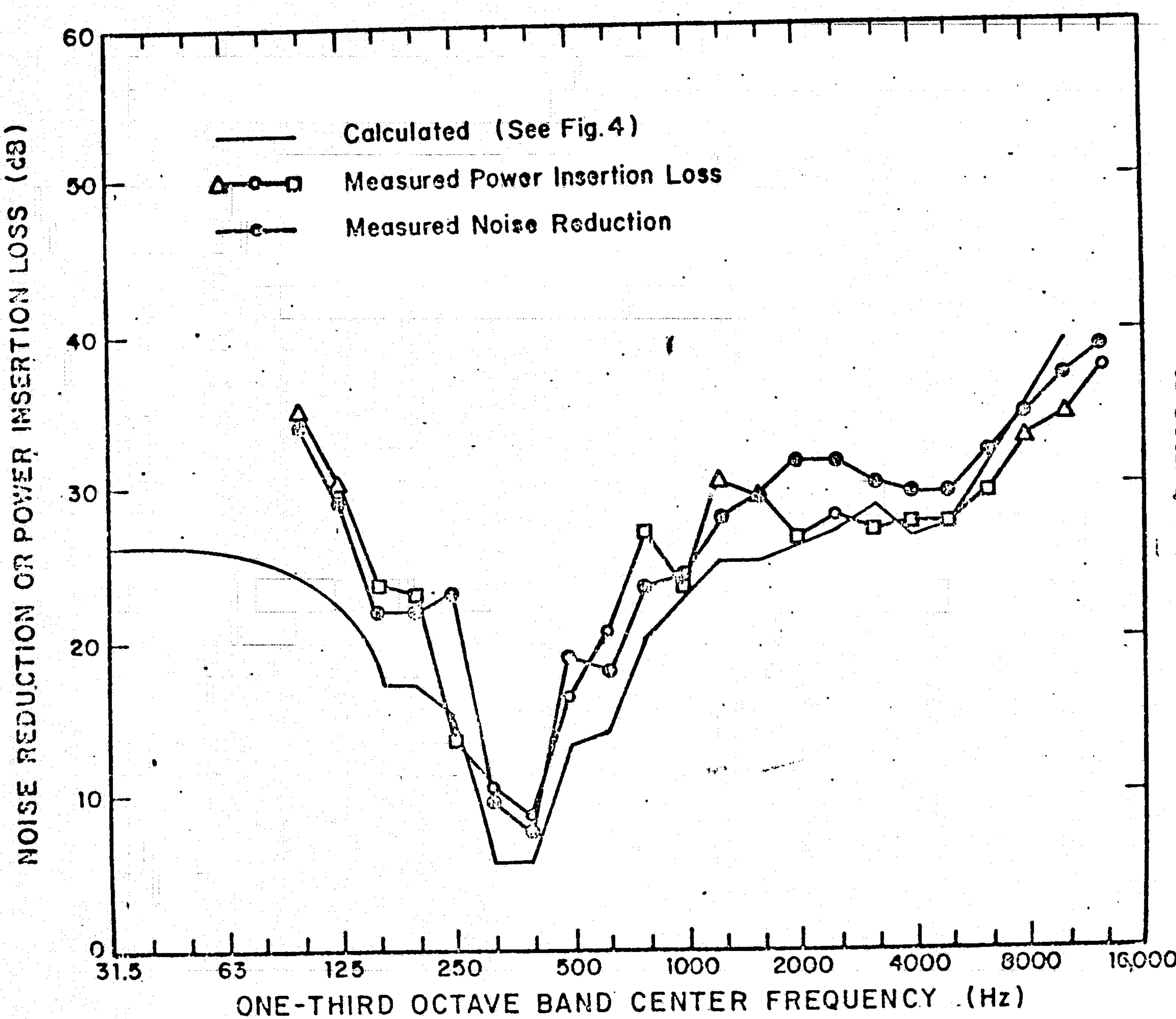

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FIG. 7. CALCULATED VS MEASURED PERFORMANCE OF THE ENCLOSURE WITH 1/2-IN. ABSORPTIVE LINING

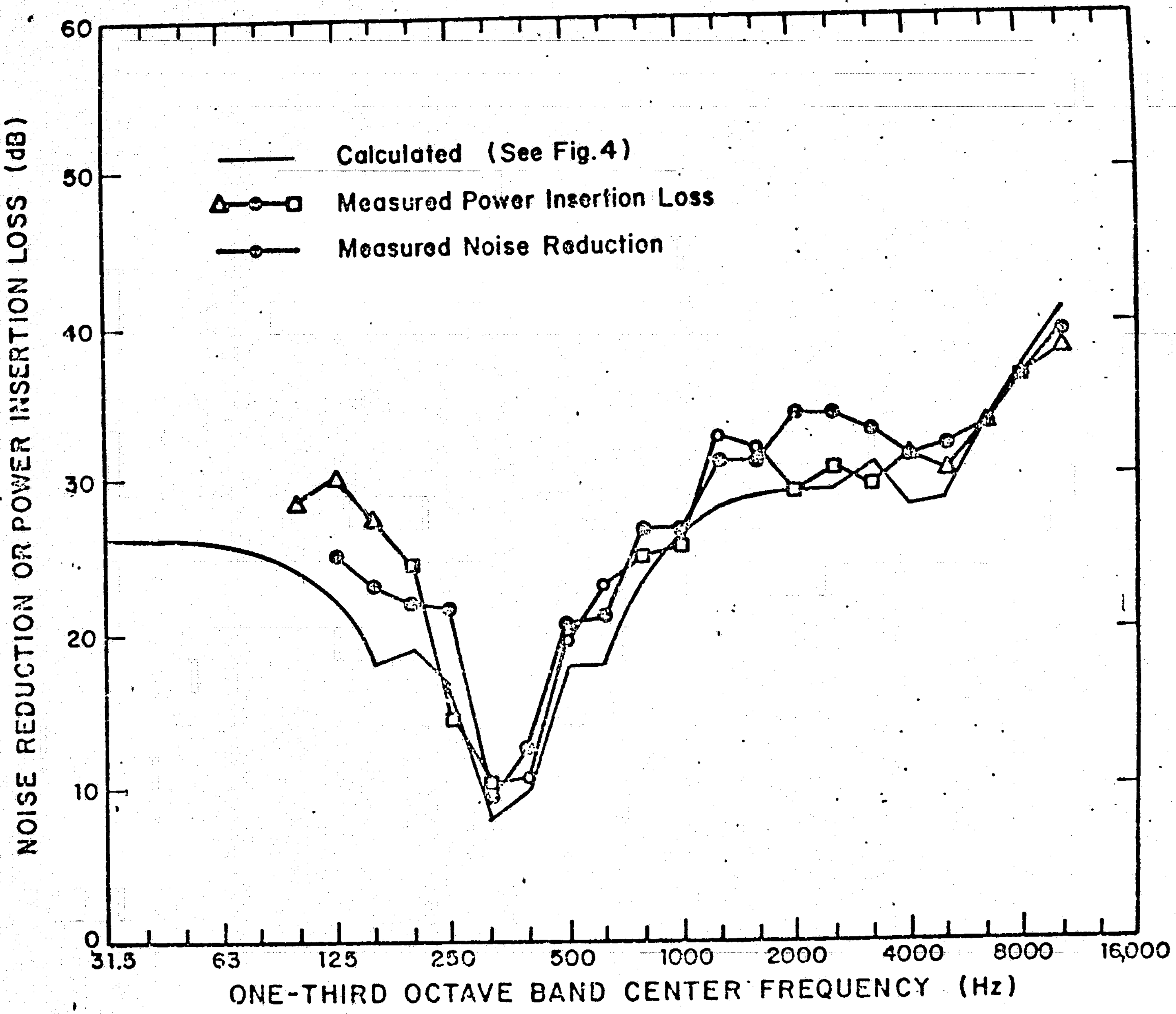

FIG. 8. CALCULATED VS MEASURED ENCLOSURE PERFORMANCE WITH 1-IN. ABSORPTIVE LINING

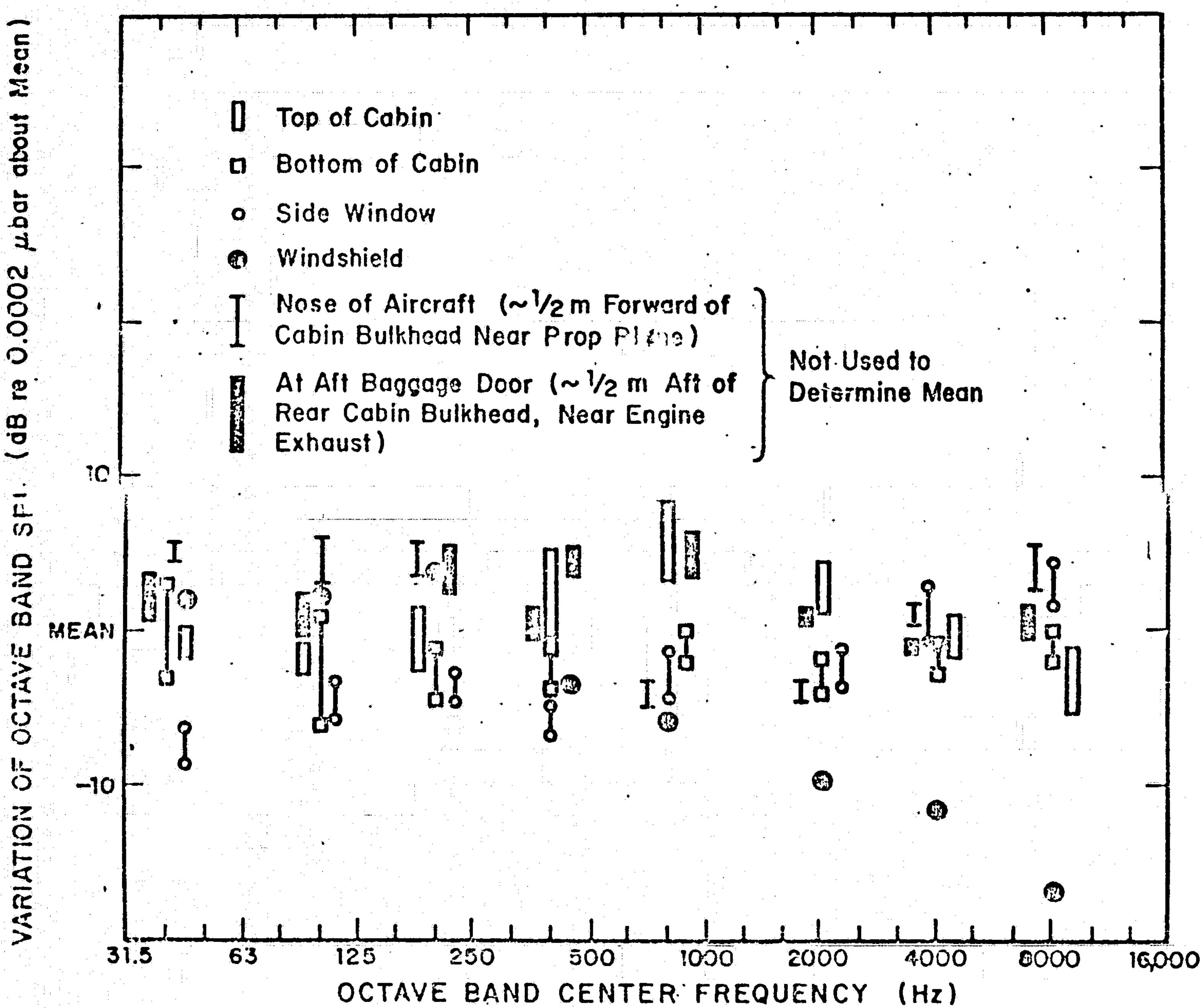

FIG. 9. VARIATION OF EXTERIOR SPL ON A LIGHT AIRCRAFT AT NORMAL CRUISE, 10,000 FT

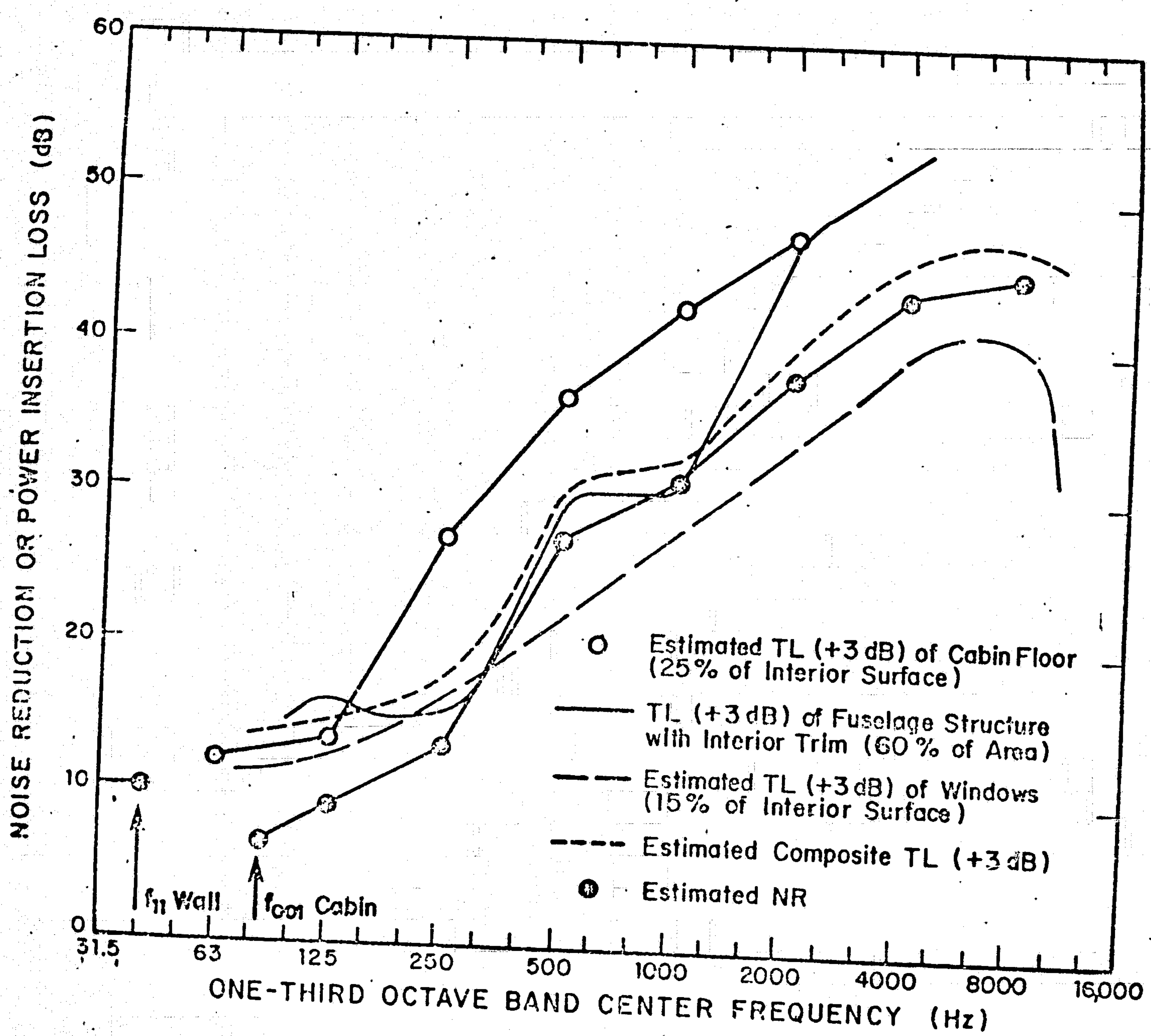

FIG. 10. ESTIMATE OF TL AND NR OF SAMPLE LIGHT AIRCRAFT

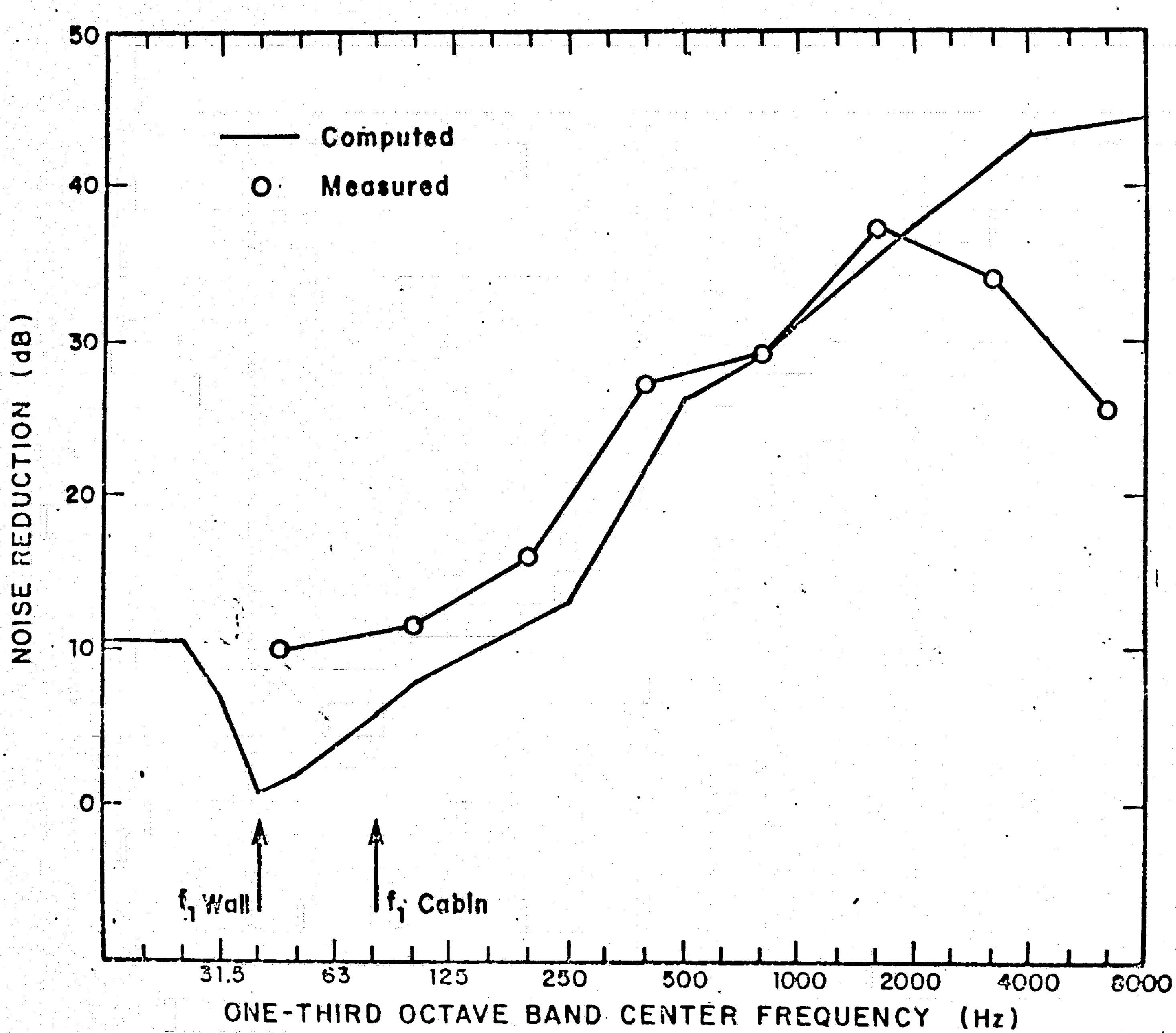

FIG. 11. COMPARISON OF MEASURED AND COMPUTED NR OF LIGHT AIRCRAFT WHOSE TYPICAL CABIN CONSTRUCTION CONSISTS OF 0.025 ALUMINUM EXTERIOR SKIN, DAMPING TAPE, 1-1/4-IN. GLASS FIBER FILLED AIRSPACE, [illegible]

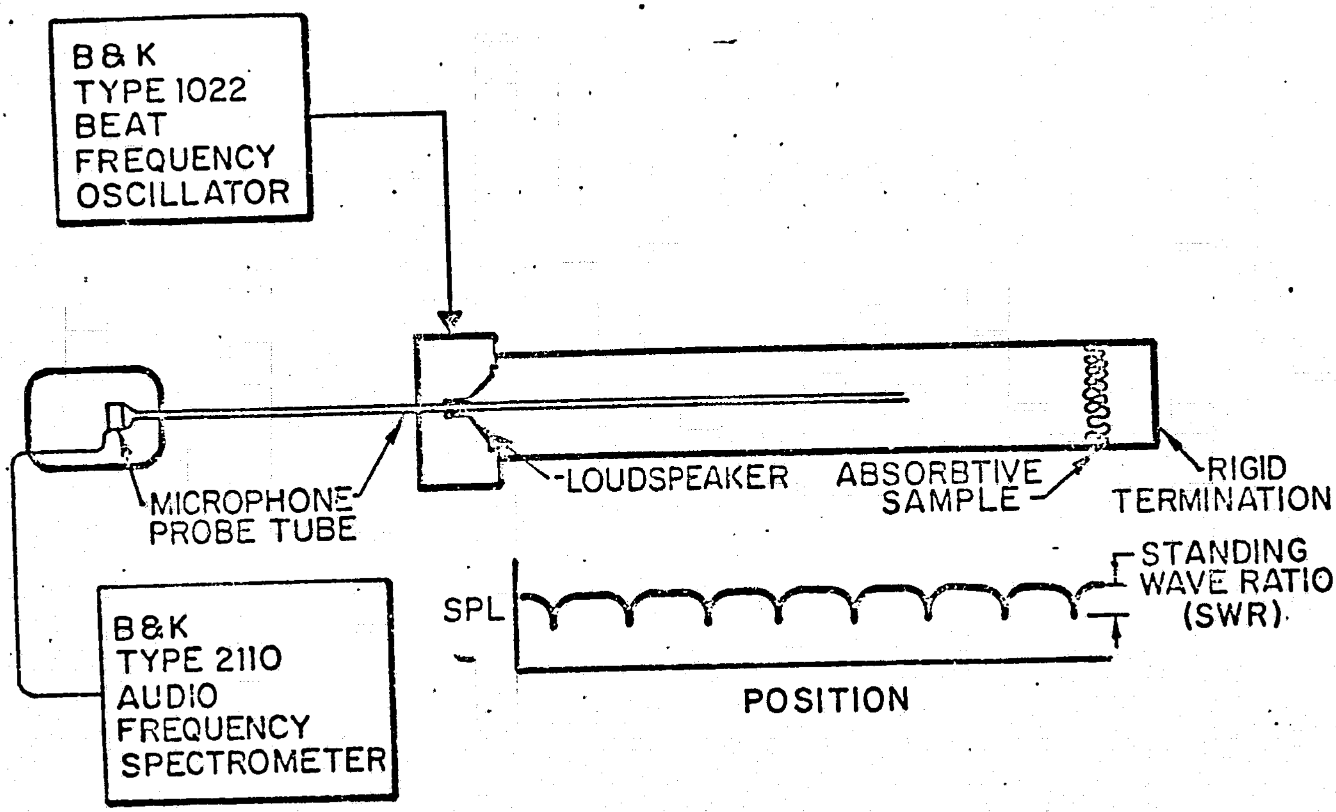

FIG. 12. SCHEMATIC DIAGRAM OF STANDING WAVE TUBE APPARATUS (B&K TYPE 4002)

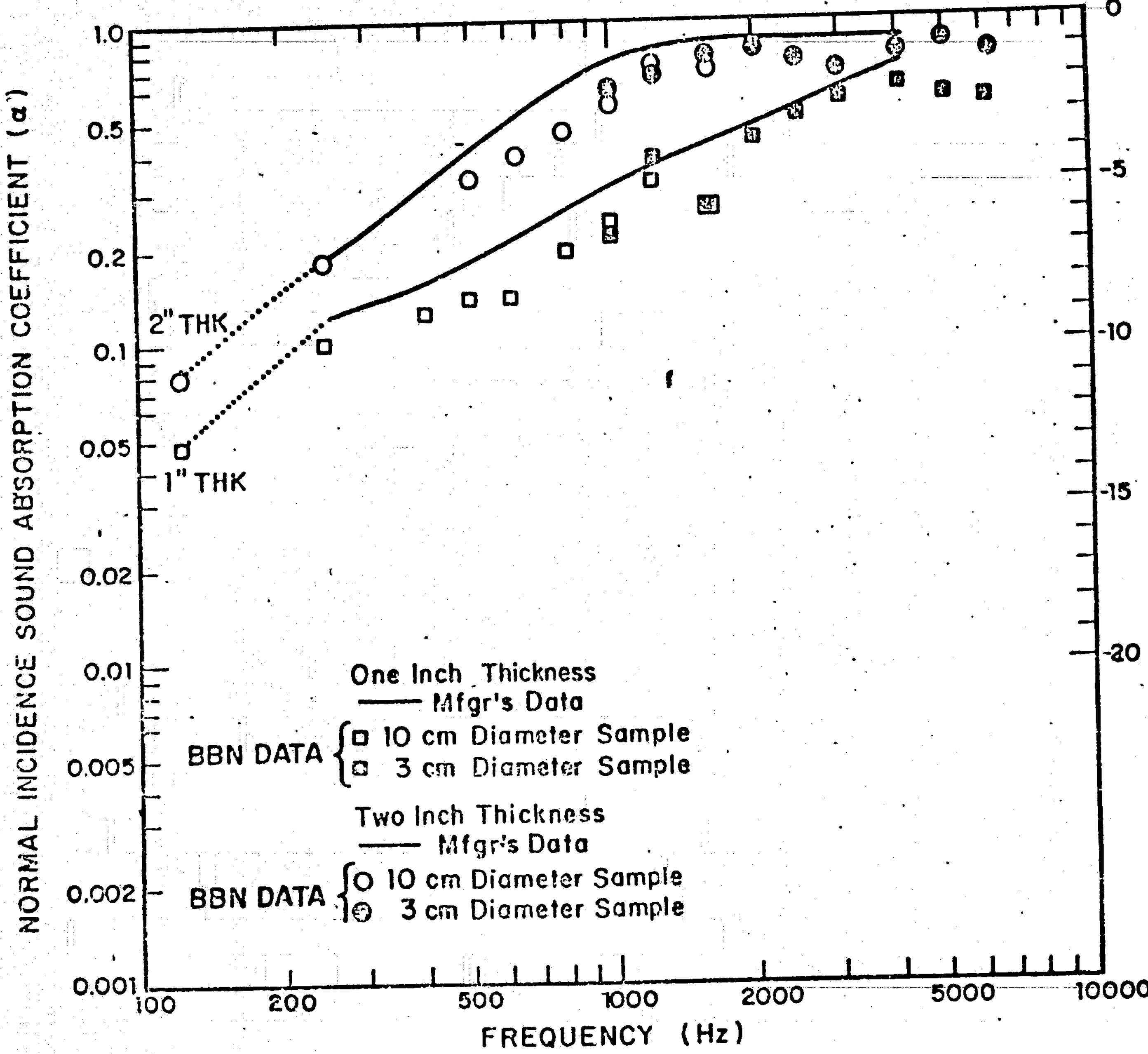

FIG. 13. ABSORPTION OF "SCOTTFOAM" 80 PPI, 1-IN. × 2-IN. THICK, VS FREQUENCY

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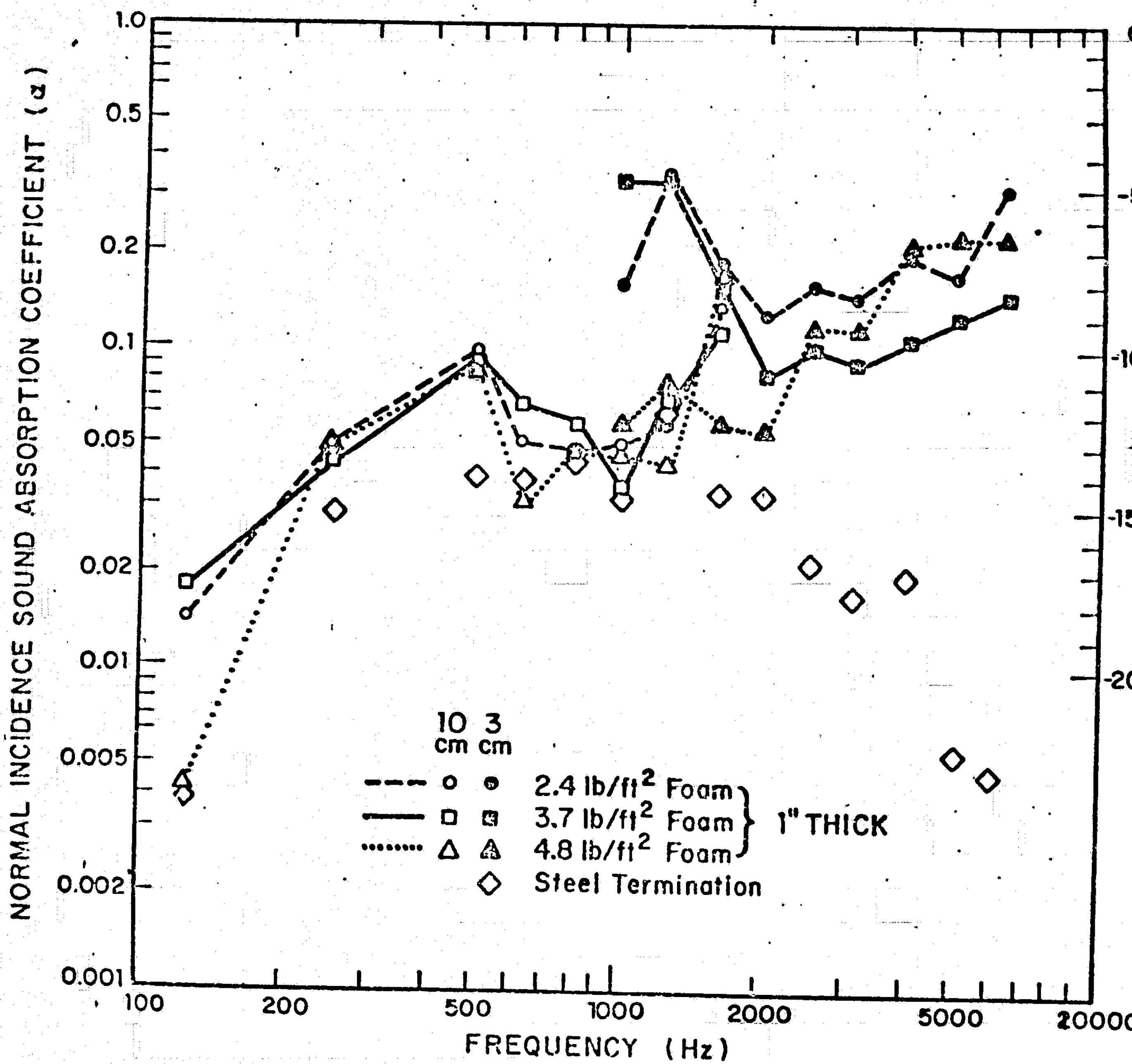

FIG. 14. ABSORPTION COEFFICIENT OF NASA FOAM

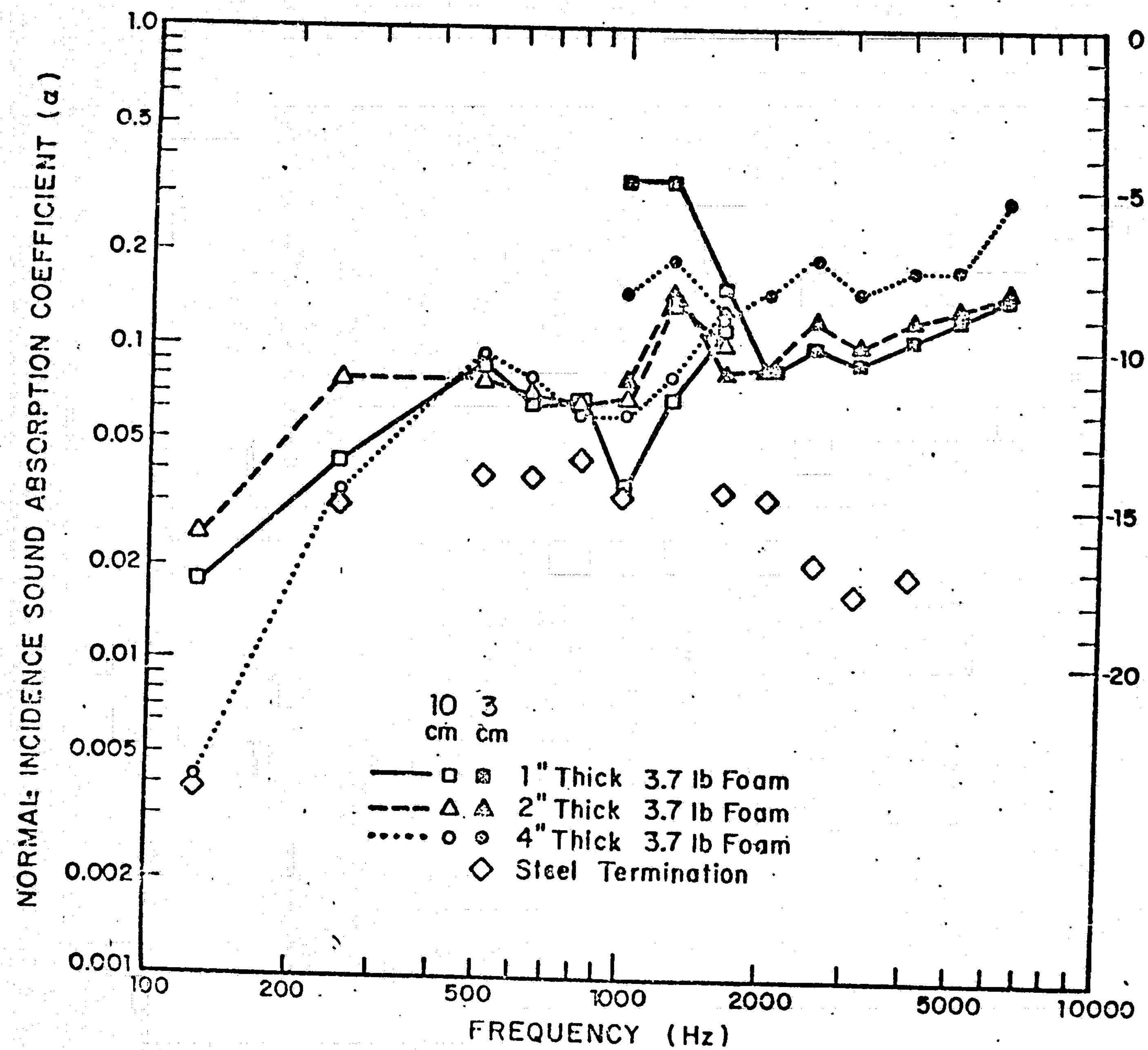

FIG. 15. ABSORPTION COEFFICIENT OF NASA FOAM

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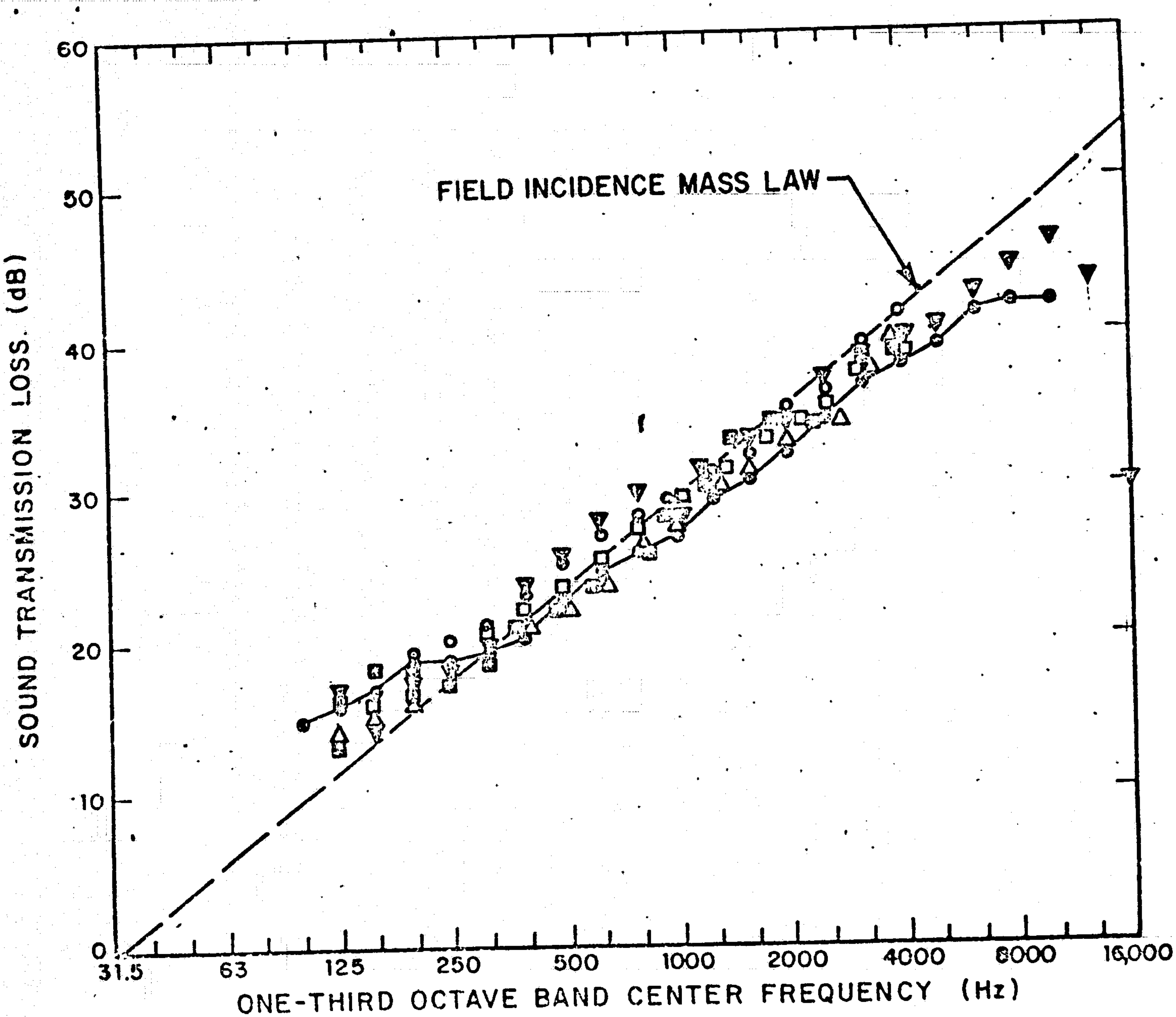

FIG. 16. TRANSMISSION LOSS OF 22 ga STEEL COMPARING RESULTS OF FIVE OTHER LABORATORIES WITH THIS LAB (—•—)

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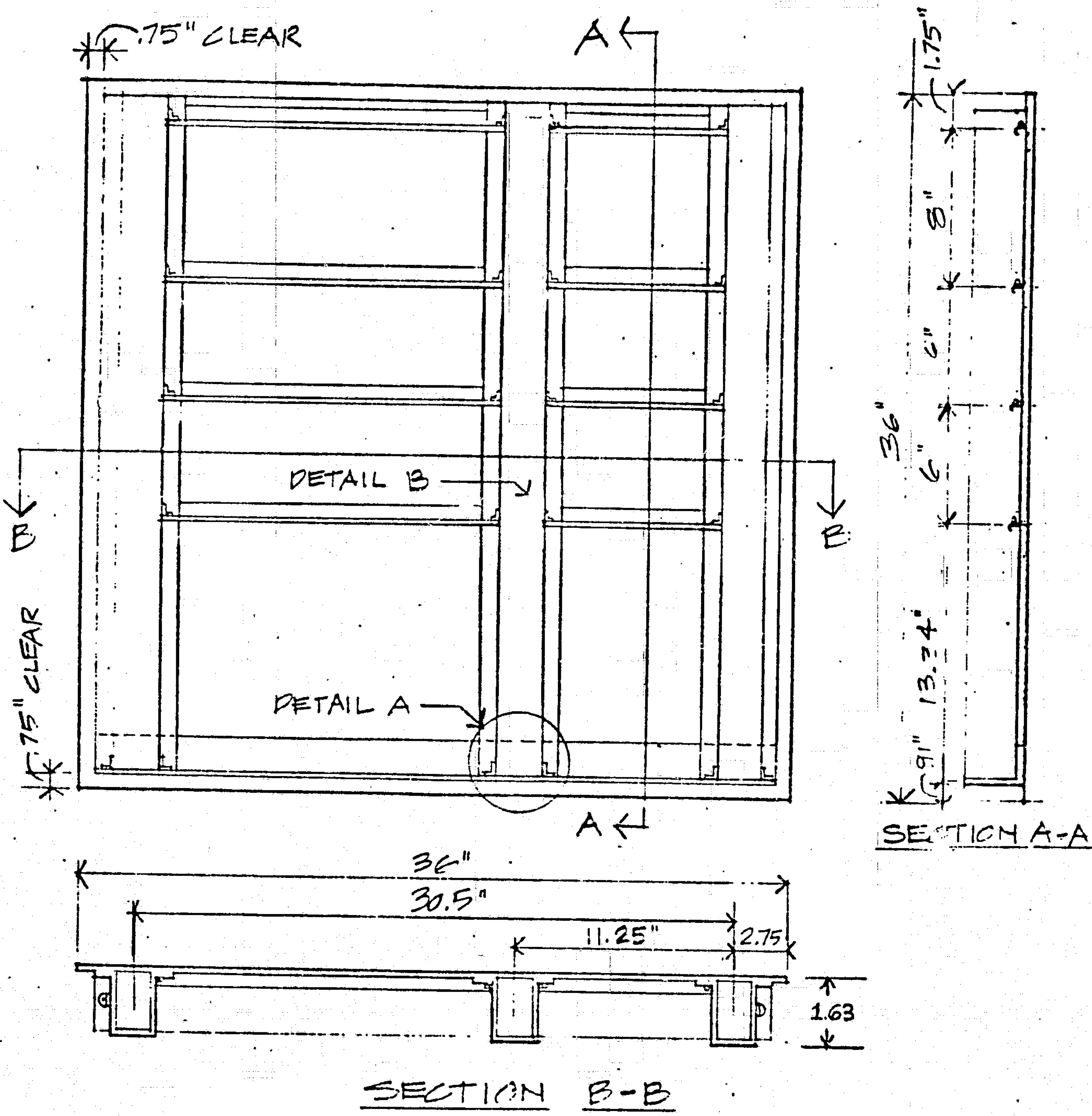

NOTES:
1. Dimensions for Stiffener Spacing are to Center Lines
2. All Flat Stock T2024 AL .032 Thick
   All Formed Stock 5052 AL .032 Thick
3. Rivit 1.5" O.C. All Mating Surfaces
4. All Rivits Airtight

FIG. 17a. NASA PANEL; SCALE 1.5 IN. = 1 FT

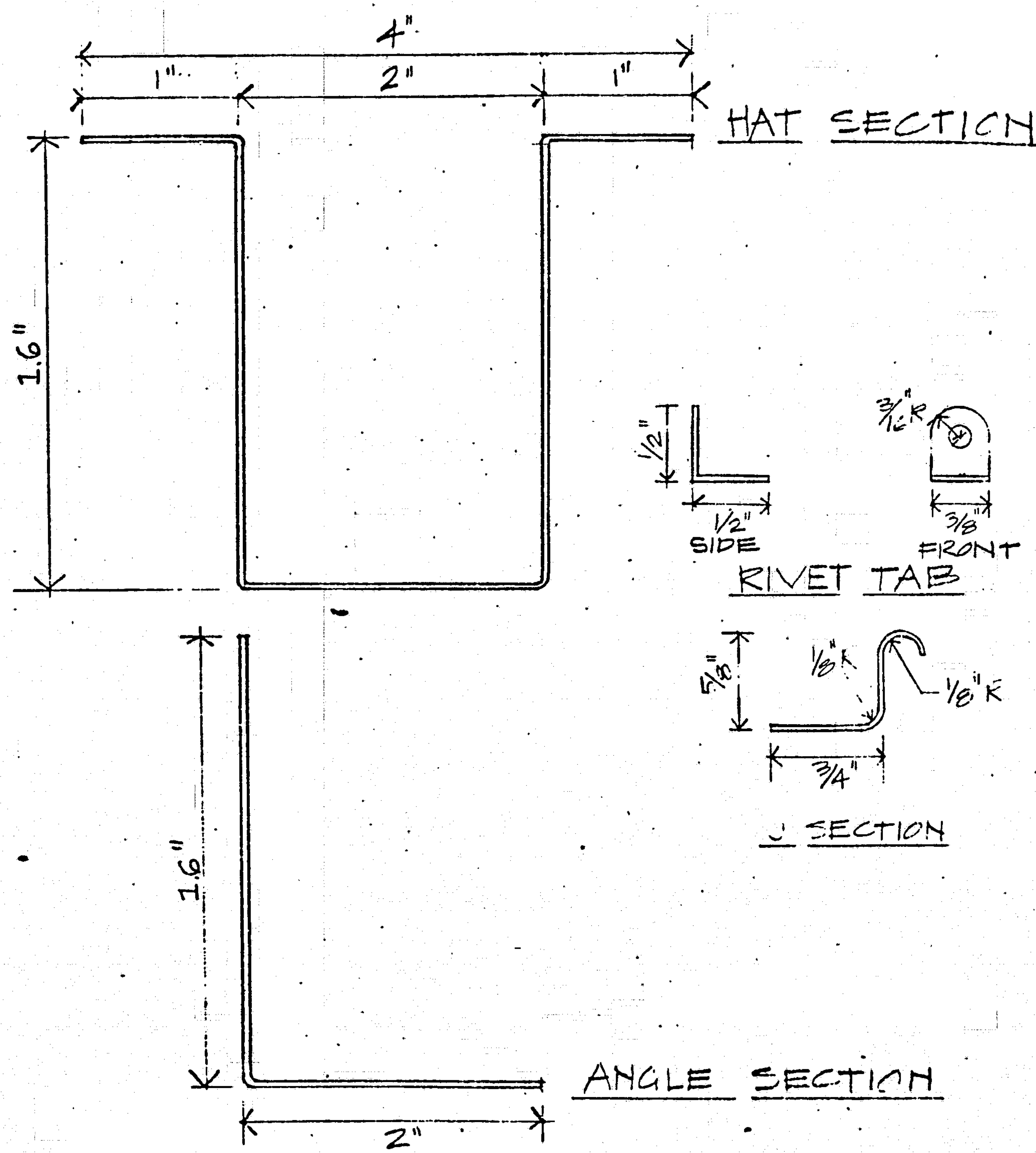

FIG. 17b. NASA PANEL DETAILS; FULL SCALE

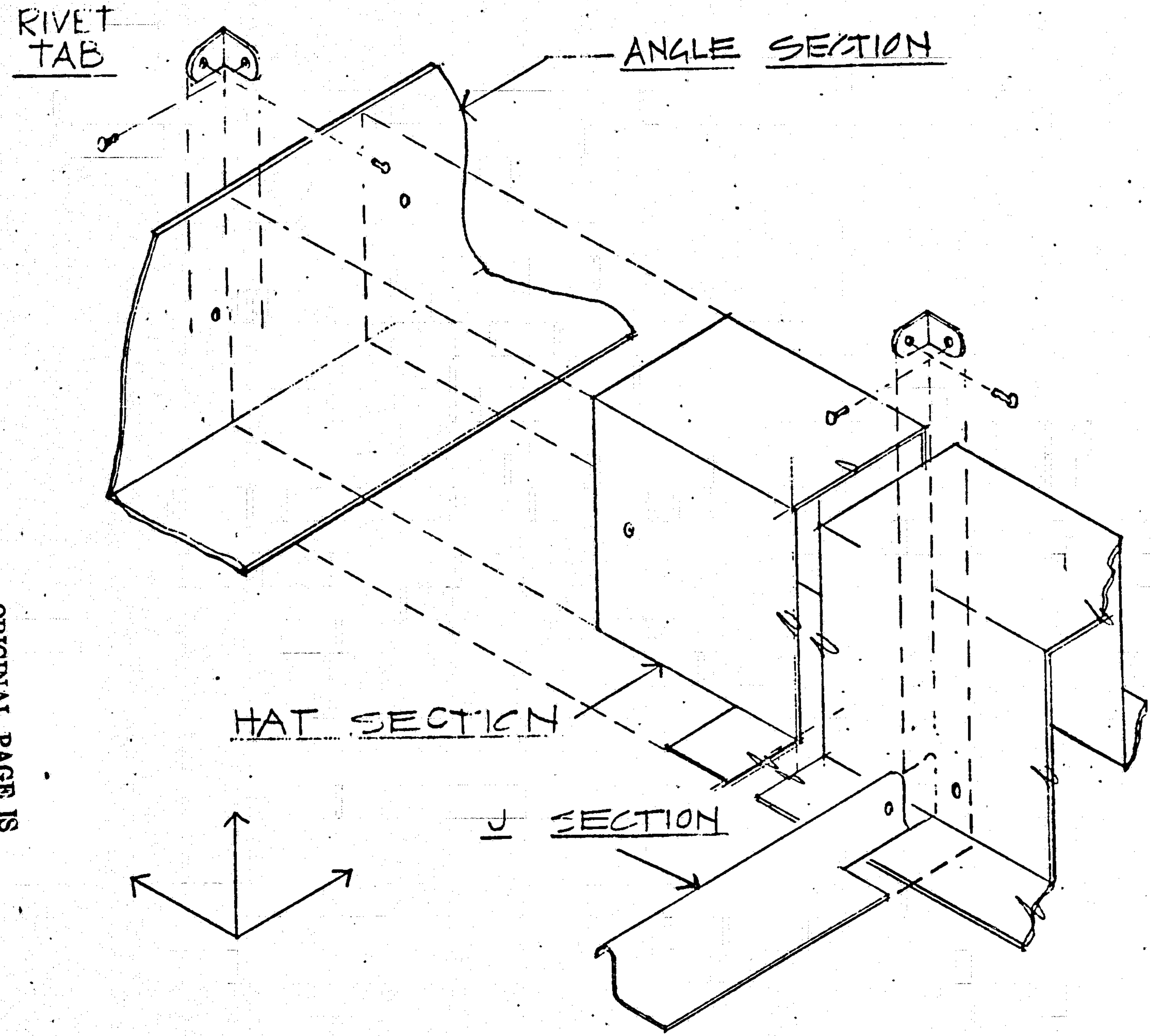

FIG. 17c. DETAIL A AND DETAIL B ASSEMBLY OF STRUCTURAL SECTIONS. NASA PANEL; SCALE: 1/2 FULL SIZE

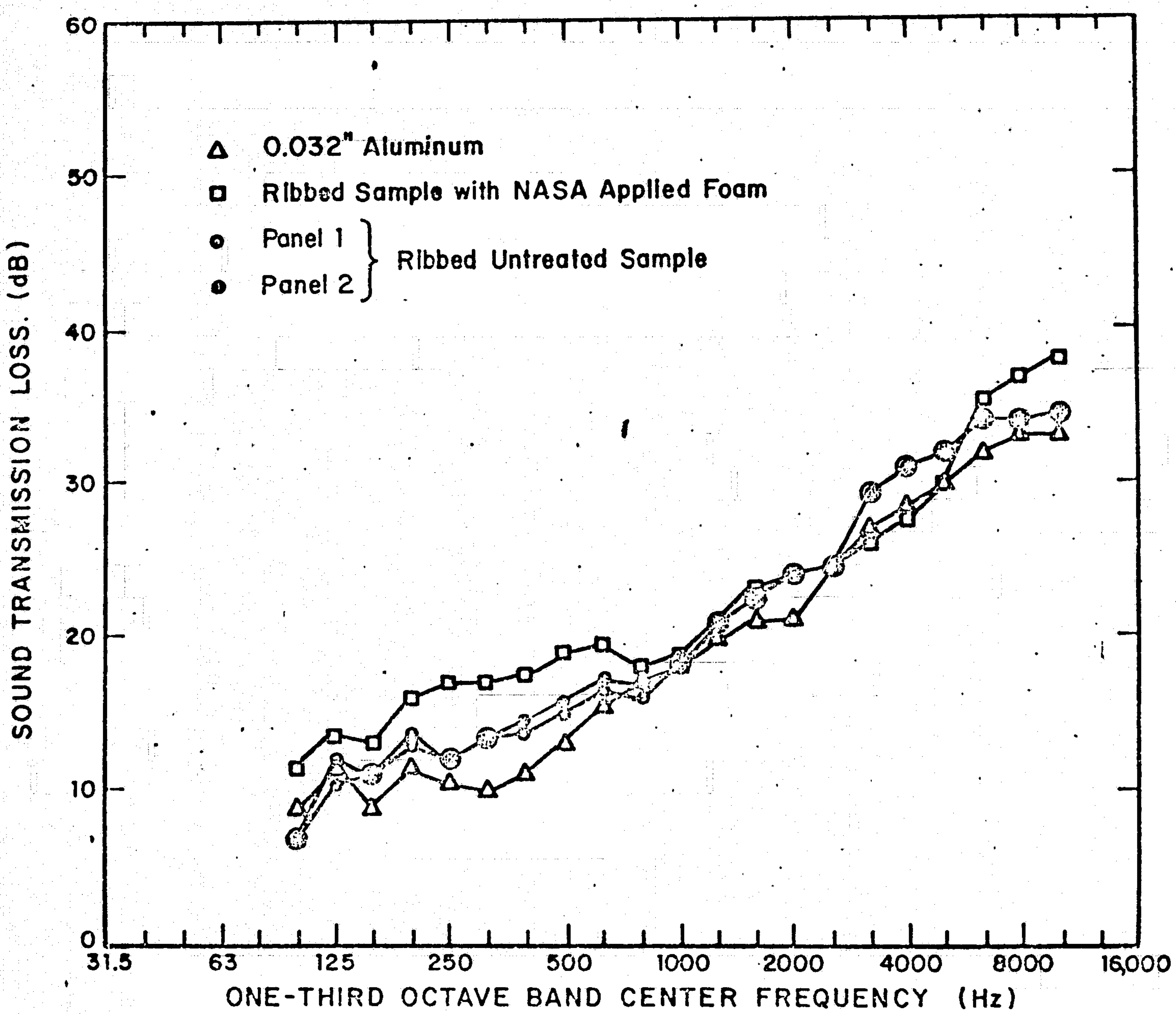

FIG. 17d. MEASURED TRANSMISSION LOSS OF SAMPLES 1 THROUGH 4

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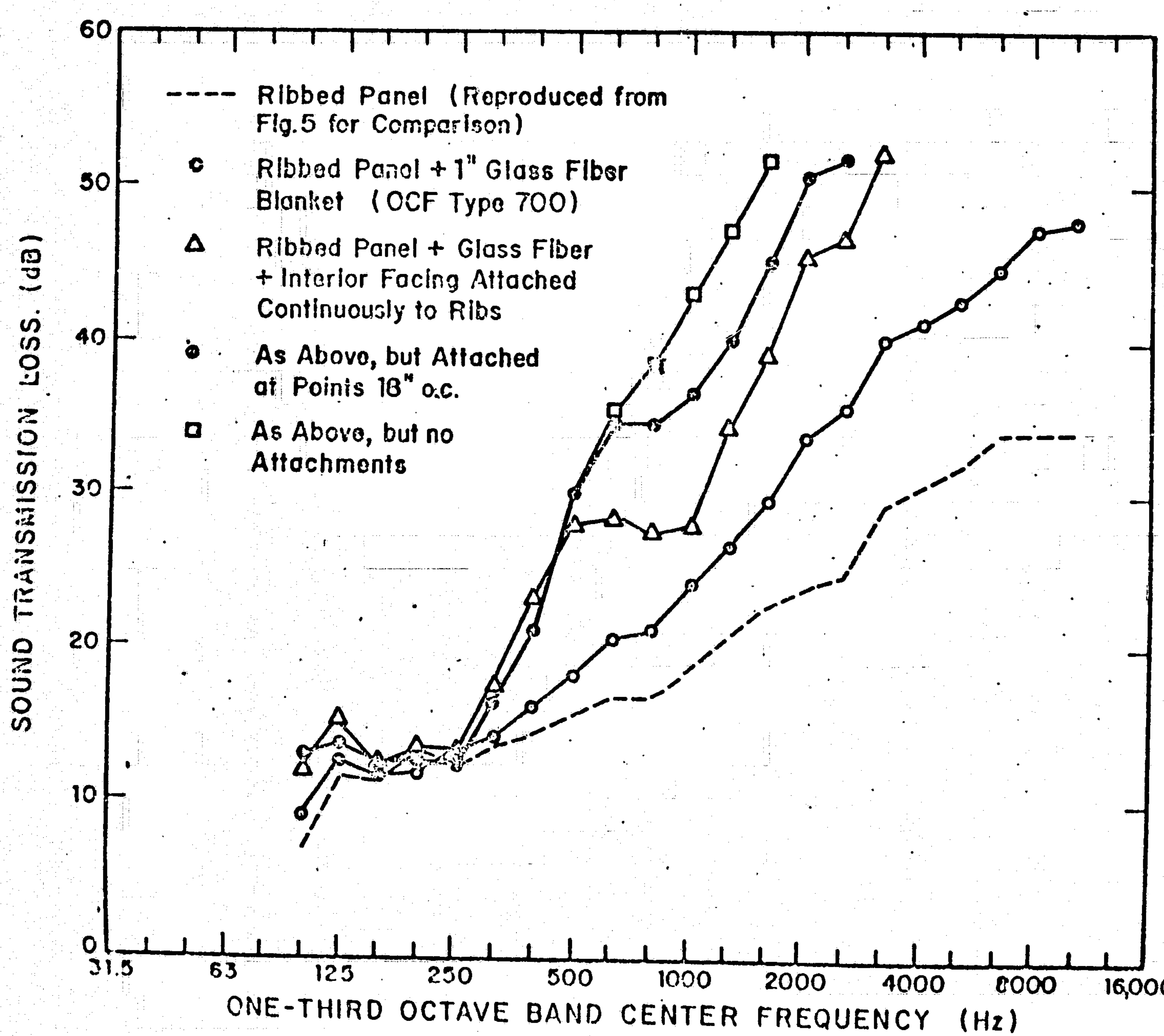

FIG. 18. TRANSMISSION LOSS OF RIBBED PANEL WITH VARIOUS SIMULATED "INTERIOR TRIMS"

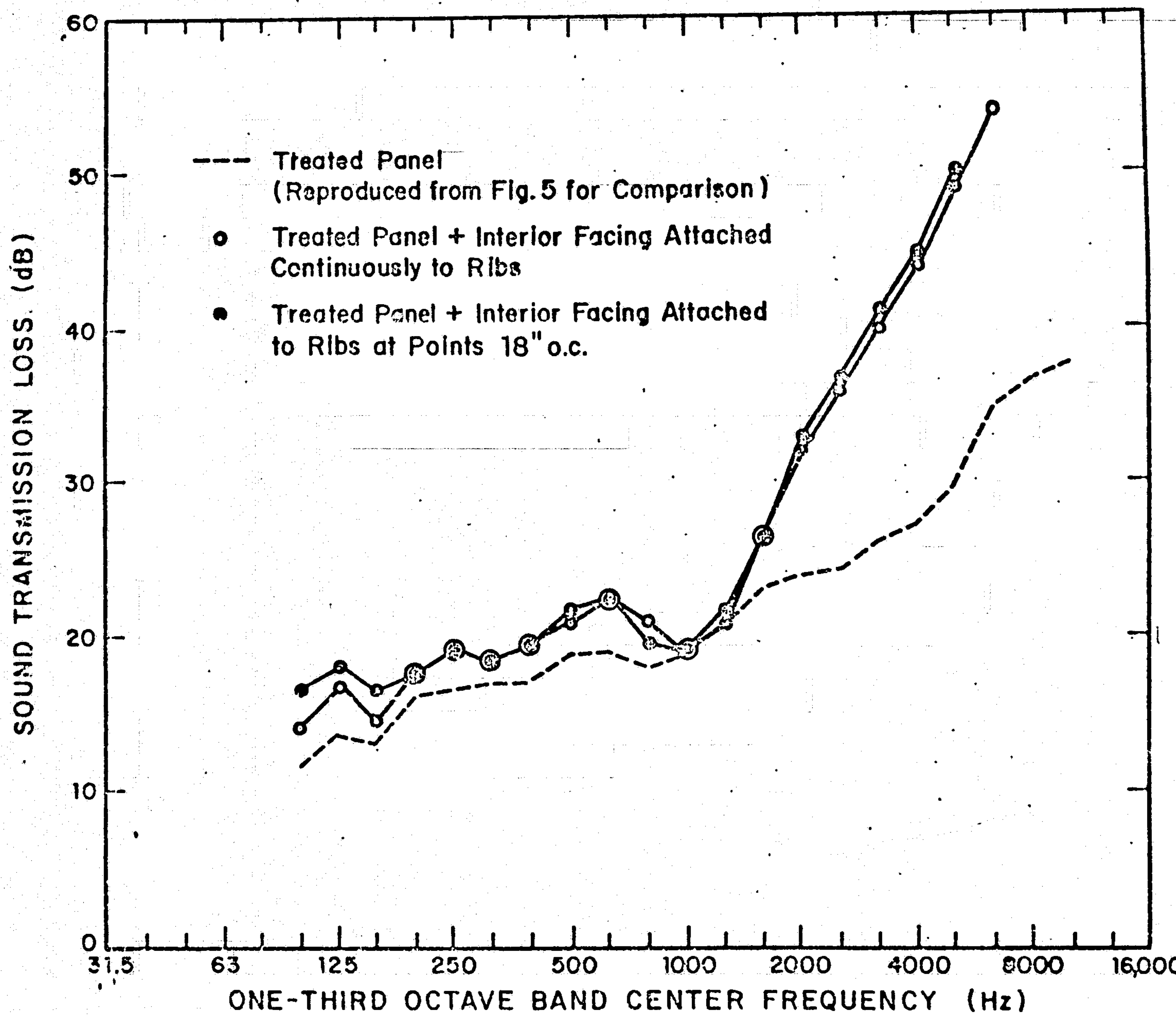

FIG. 19. TRANSMISSION LOSS OF NASA TREATED PANEL WITH "INTERIOR TRIM"

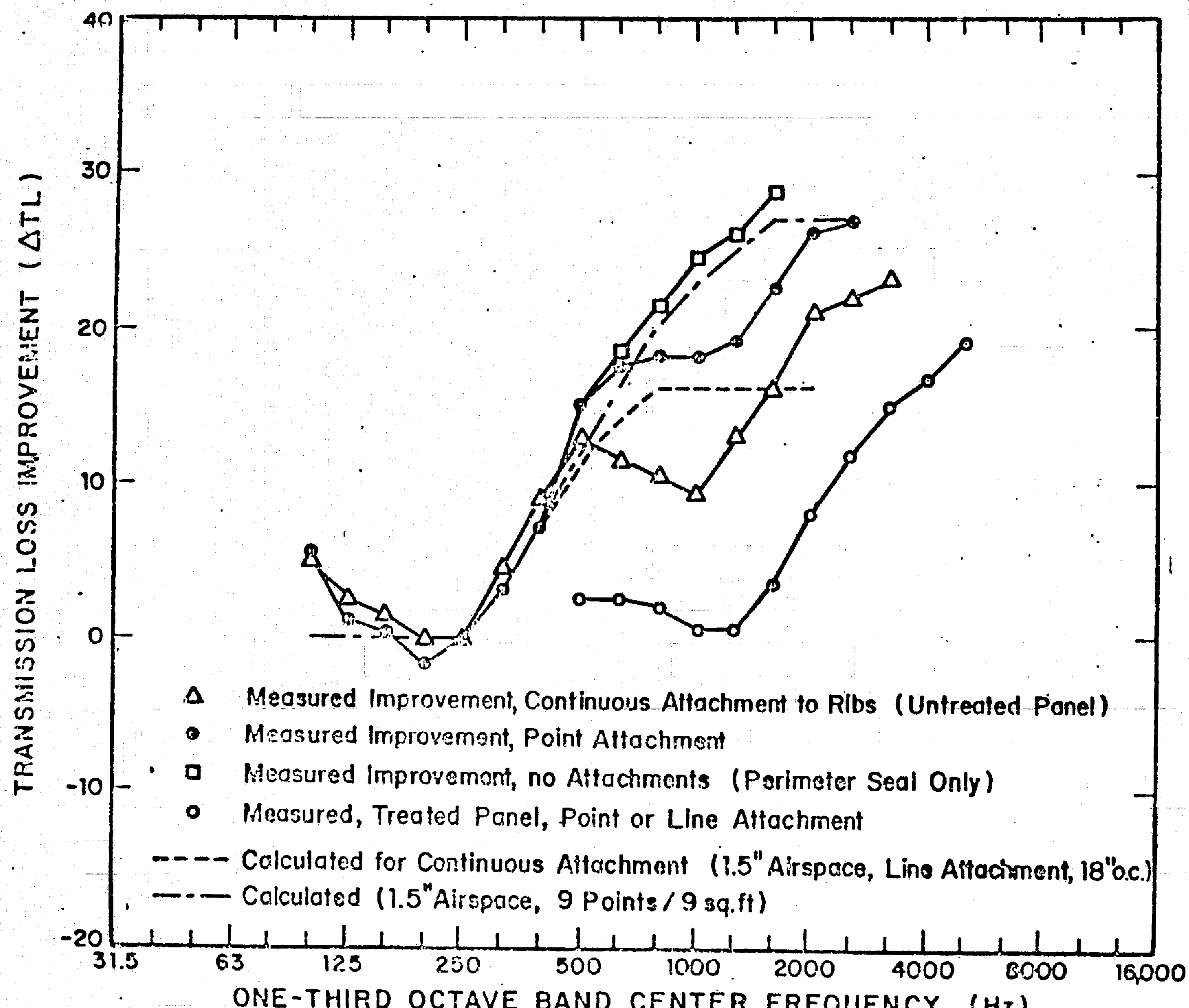

FIG. 20. IMPROVEMENT IN TL WITH THE ADDITION OF SIMULATED "INTERIOR TRIM"

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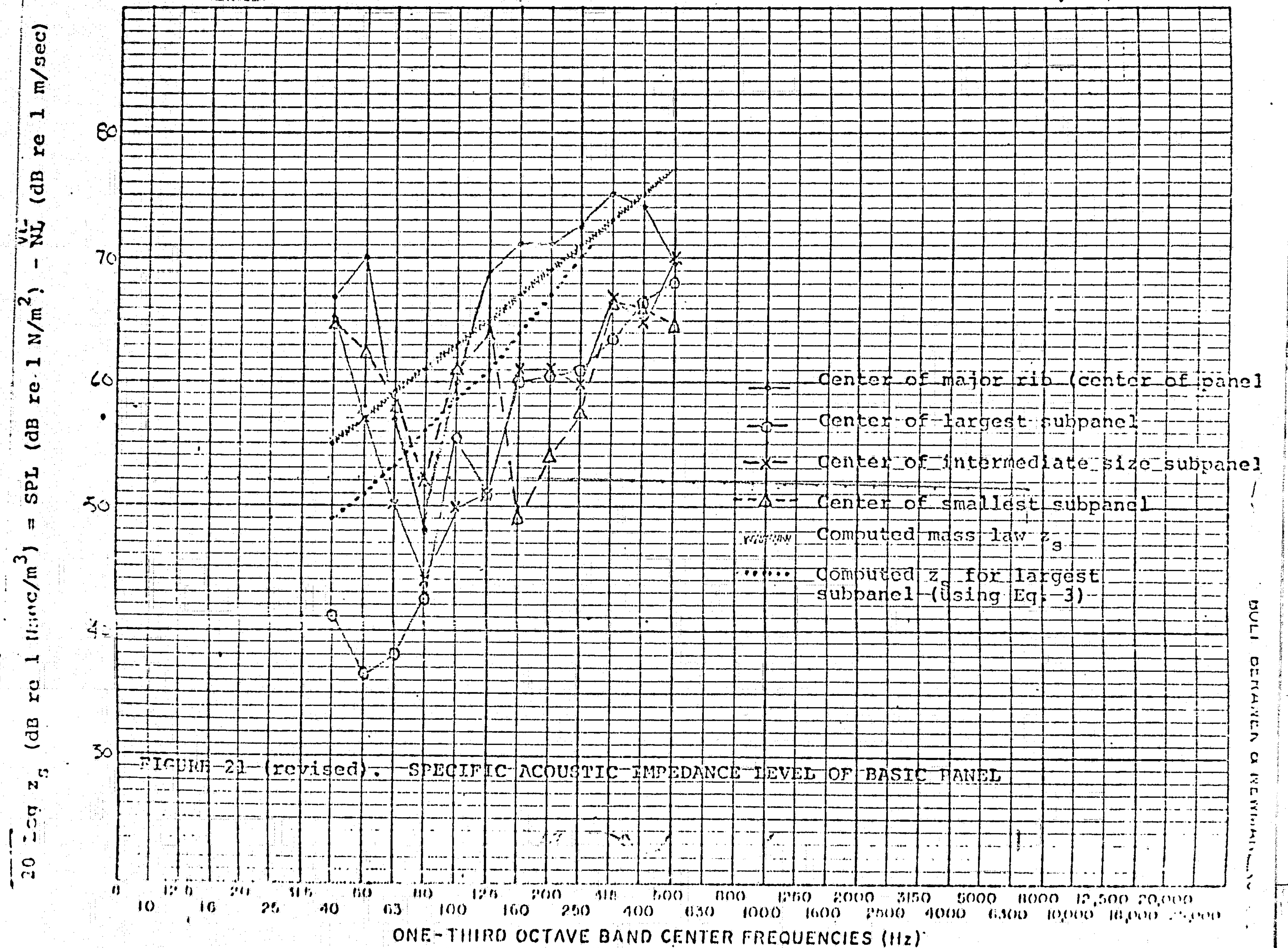

FIGURE 21 (revised). SPECIFIC ACOUSTIC IMPEDANCE LEVEL OF BASIC PANEL

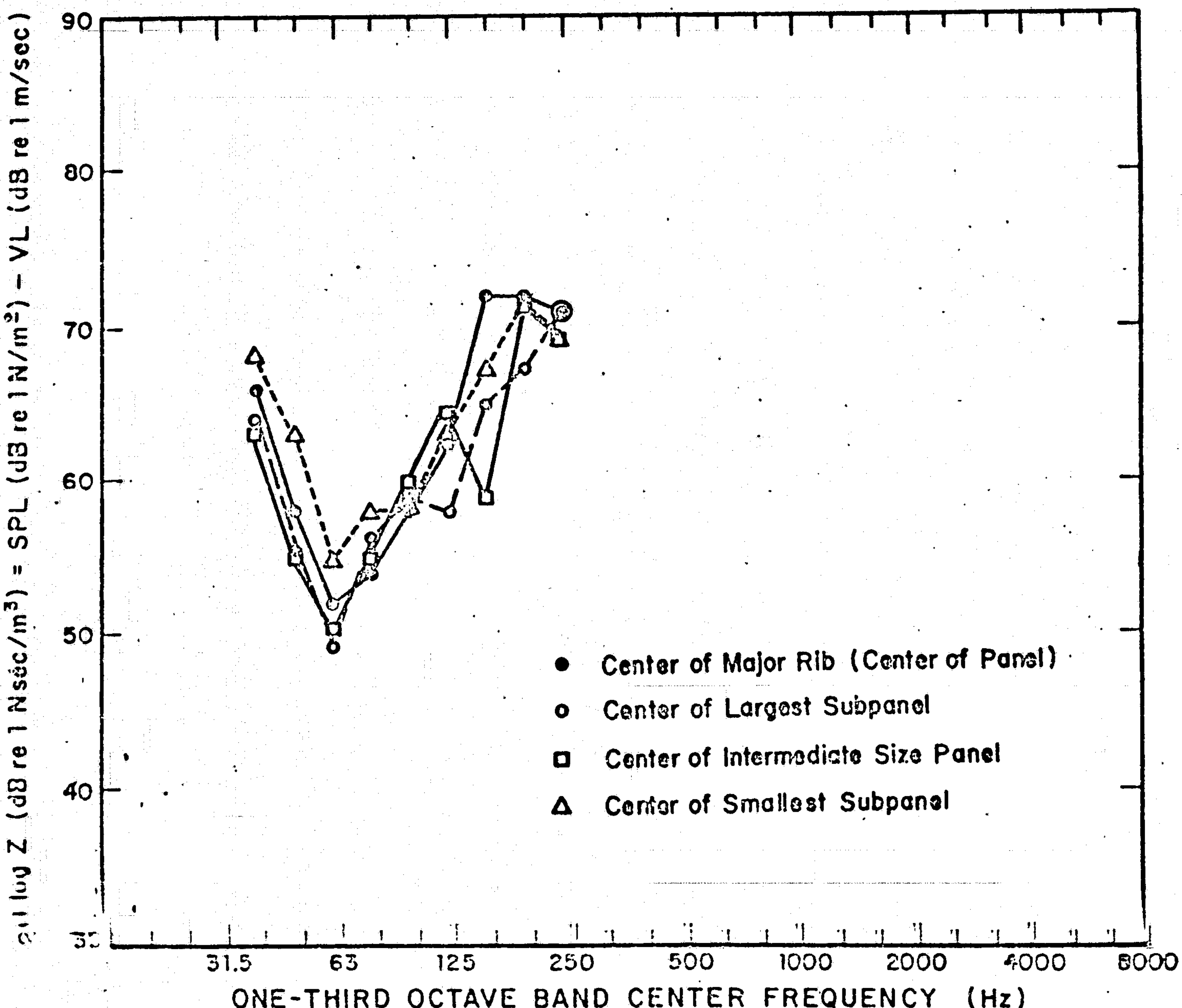

FIG. 22. ANALOGOUS IMPEDANCE LEVEL FOR NASA TREATED PANEL

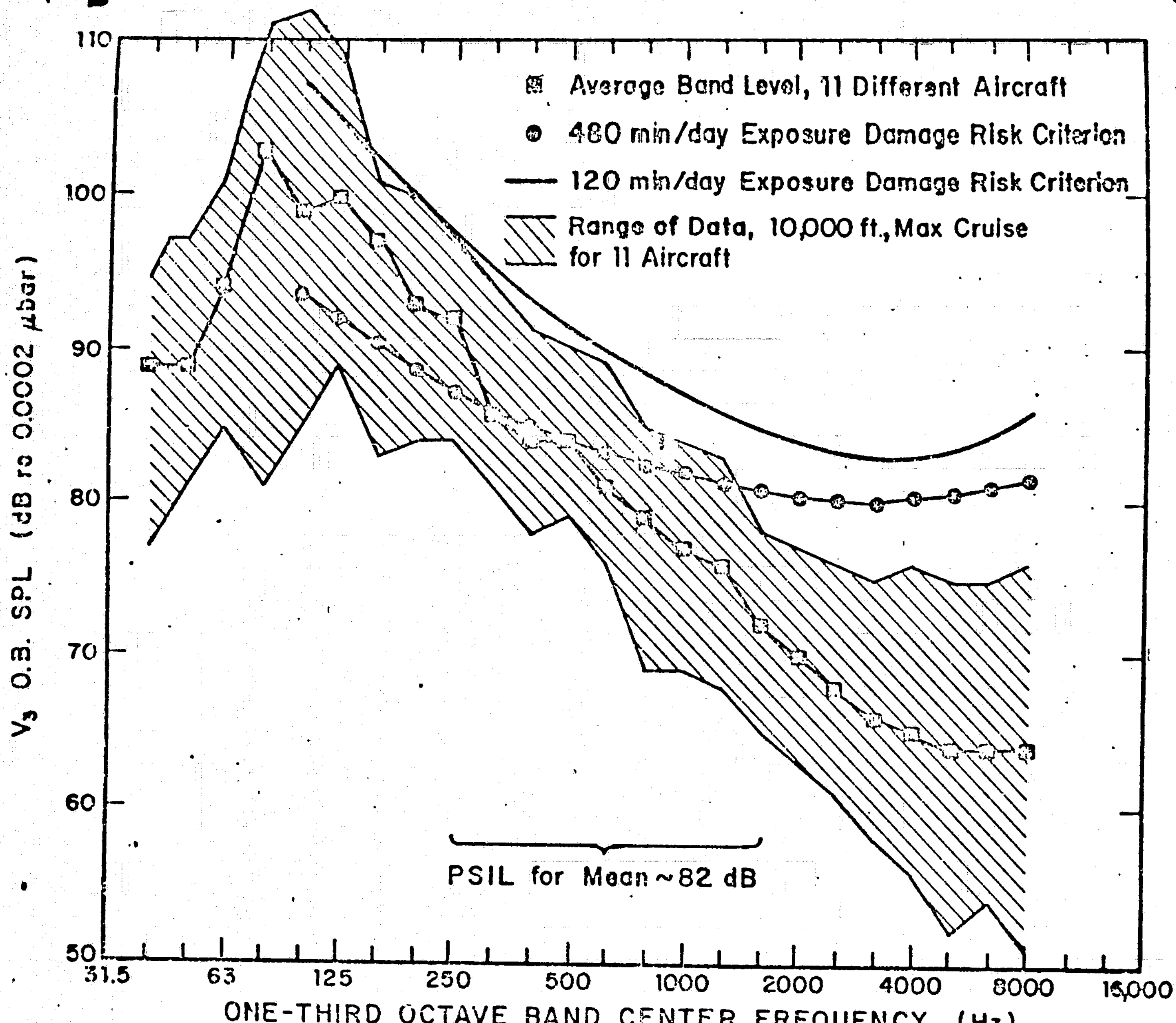

FIG. 23. MEAN AND RANGE OF SPL IN 11 LIGHT TWIN AIRCRAFT COMPARED WITH HEARING DAMAGE RISK CRITERIA FOR CONTINUOUS EXPOSURE (AFTER TOBIAS)

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